MANEUVERABLE PENETRATION SYSTEM FOR HORIZONTAL EXPLORATION IN SOFT GROUND

[pt.2]

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by

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CHAPTER 5

CONCLUSION

Two basic horizontal maneuverable penetration systems(MPS) are presently available, which can be modified for soft ground exploration. The mandrel MPS consists of a Dyna-Drill downhole hydraulic motor, bent or articulated sub, drill pipe, a conventional diamond or tricone bit, and various miscellaneous surface support equipment. The thrust applicator MPS is built around a DRILCO thrust applicator and includes: a Century Electric or a W. H. Nichols hydraulic drill motor; a conventional drag or tricone drill bit; a CONOCO deflection shoe, orientating motor, and downhole hydraulic valving system; and the necessary embilical cables interconnecting the thrust applicator with the required surface support systems. Both of the MPS's contain module spaces for an electronic navigation package and various geotechnical and geophysical sensing devices.

Within the framework of these two basic approaches four MPS's have been proposed to operate in four selected urban environments.

Tables 3.8 and 4.1 are repeated here for convenience and will be referred to as Tables 5.1 and 5.2, respectively, throughout the following discussion. Selection A and B, in Table 5.1, will operate well in a stiff clay or dense sand out to 1600 ft(488 m) horizontally, while only Selection A will extend out to 700 ft(214 m) in soft clay or loose sand. In order to drill a greater distance a washover drill pipe is required to reduce the soil friction. Selection B's own weight hinders its directional control capability in soft clay. Both of these mandrel MPS's would be expected to perform satisfactorily in a residual soil if pebble size particles did not clog the drill bit or lodge between the drill body and hole wall. The minimum, continuous radius of curvature for the mandrel MPS is associated with a build angle of 12°/100 ft in stiff clay or dense sand while in soft clay or loose sand it drops to 9°/100 ft as shown in Table 5.2. The maximum "kink" radius of curvature for the mandrel MPS is measured at an associated build angle of 26°/100 ft.

The 5-3/4 in(14.6 cm) DRILCO thrust applicator, with modified anchor pads (1-1/2 x 8 in(3.8 x 20.3 cm)) was basic to the two thrust applicator MPS's which were analyzed as Selections C and D in Table 5.1. Either the Century Electric motor or the W. H. Nichols

Table 5.1 Final Design Selections

Selec'n	A	B	C	D
Drill Motor	Dyna-Drill	Dyna-Drill	Nichols Hyd. Motor	Century Electric Motor
Drill Motor O.D. (in)	2-3/8	6-1/2	5	3-11/16
Length (ft)	7	19.6	4	4.5
Normal Force Device (NFD)	Drill Pipe	Thrust Appl. Drill Pipe	Thrust Appl.	Thrust Appl.
NFD O.D. (in)	2-3/8	8(T.A.) 4-1/2 (D.P.)	5-3/4 or 8	5-3/4 or 8
Direction Control	Bent Sub	Def.Shoe (T.A.) Bent Sub (D.P.)	Def. Shoe	Def. Shoe
Bit Type	Diamond or Drag	Tricone	Tricone	Tricone
Hole Dia. (in)	4-1/2	12	7	7
Comments	Excellent annulus size low flow rate, high torque	Max. annular size, max. torque RPM, high flow rate can be a problem	Opt. annular space, short length high torque low flow	Opt. annular space, min. flow requirements, short length, problem w/shorting

Table 5.2 Radius of Curvature for the MPS

	Radius of Curvature in Stiff Clay, Dense Sand (System limited) (ft)	Radius of Curvature in Soft Clay, Loose Sand (Formation limited) (ft)	Minimum Horizontal Distance (Vertical Entry) (ft)		Comments
			Stiff	Soft	
Mandrel	1145(5^o/100') to 216(26^o/100') Opt: 475(12^o/100')	1145(5^o/100') to 570(20^o/100') Opt: 635(90^o/100')	475	647	1) The system limit is based on the wear factor during bending of the rubber stator in the Dyna-Drill.
Thrust Appl.	1145(5^o/100') to 380(15^o/100') Opt: 715(8^o/100')	1145(5^o/100') to 380(15^o/100') Opt: 715(8^o/100')	715		1) The system limit is based on the minimum allowable deflection from bending which the thruster cylinder splime can withstand. 2) The soft ground limits are based on the ability of the thruster to develop thrust in this environment.

hydraulic motor perform equally well as drilling motors. The thrust applicator MPS was found to theoretically perform well in a stiff clay or dense sand while being capable of operating at depths up to 500 ft(153 m) and out to a horizontal distance of 5000 ft(1525 m). However, based on theoretical calculations and field experience, the presently configured thrust applicator MPS will not be able to develop the necessary shear resistance at the anchor pad-soil interface in soft clay (i.e. less than 1.0 tsf unconfined compressive strength) in order to penetrate at any depth for any horizontal distance. The thrust applicator MPS can operate in a residual environment with the same limitations as the mandrel MPS. In addition, this MPS has the directional control ability to avoid objects using a minimum radius of curvature with a build angle of 8°/100 ft while its "kink" radius of curvature is 15°/100 ft. These values are less than the mandrel MPS because of the rigidly connected front section on the thrust applicator MPS.

These two basic MPS's are presently available and have been tested in several different soil conditions. However, prior to the investigation for this thesis, the thrust applicator system was thought to be a conceptual model only.

In addition to the preliminary MPS design, three other important conclusions were reached during this investigation. First, the difficulty of drilling a horizontal hole is dependent upon the control of the drilling mud recirculation system. The proper bentonite drilling mud could provide enough lubricity to significantly reduce the skin friction between the soil and drill steel. The drilling mud also provides hole stability while cleaning the annular space of drilling fines, preferably without hydraulically fracturing the soil.

Secondly, the soil friction effect could be reduced with a neutrally buoyant drill pipe or thruster cable. If either the drill pipe or cable were neutrally buoyant in the horizontal section (Section II in Figure 3.14) of the drilling path, both Selection A and B could drill to at least 5000 ft (1525 m) horizontally at a depth of 500 ft(153 m).

Finally, a dimensionless analysis of the four alternate MPS's allowed their comparison to be quantifiable objective rather than subjective.

In conclusion, the two basic maneuverable penetration systems can be manufactured with the present state of technical knowledge as explained in Chapter 4. The most efficient combination of these subsystems, defined through the results of a

dimensionless analysis, will be able to penetrate a soft ground condition down to a depth of 500 ft (153 m) along a 5000 ft(1525 m) horizontal drill path.

CHAPTER 6

RECOMMENDED FUTURE RESEARCH

Throughout the course of research for this thesis, several important and specific items have been found to require future research in order to advance the state of knowledge in maneuverable, horizontal directionally controlled drilling in soft ground. Below are listed a few of the more important items. This list can obviously be expanded as research and development continue in this embryonic field.

1) The mandrel and thrust applicator MPS should be compared through a competitive field test in one of the four proposed geological environments.

2) During this test an instrumented package should be mounted on both MPS's to measure the normal force, torque, vibration, and RPM at the drill bit.

3) Conduct extensive research into the effectiveness of various drilling muds to stabilize the hole and retain the soil particles in the horizontal section of the drill hole.

4) The dimensionless parameter relationships between geology and the MPS should

be expanded and verified to allow the users to make a logical selction of subsystems in varying subsurface strata.

5) Methods of producing neutrally buoyant cable and pipe should be investigated to determine the feasibility of neutral buoyancy for reducing the skin friction along the drill pipe or cable.

6) The DRILCO thrust applicator should be redesigned for efficient soft ground operation, or possibly developing another thruster which operates on the concept of vermiculating motion such as the WORM.

7) The following subsystems should be further developed to improve existing equipment: a closed circuit downhole valving system for the DRILCO thrust applicator; a 7-8 in(17.8-20.3 cm) diameter tricone coring bit; and a more refinely controlled hydraulic system for anchor pad and deflection shoe extension.

8) Determine the applicability of the W. H. Nichols hydraulic motor to the mandrel MPS.

9) Further research should be conducted into the actual condition of the drill pipe in the bore hole to determine whether slender, free-ended column buckling failure is occurring.

10) Investigate the assumption that the annular pressure(ΔP_a) of the drilling fluid is in equilibrium with the effective stress at the bore hole wall boundary.

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APPENDIX A

LIST OF TERMS, DEFINITIONS, AND SYMBOLS

A - Arc angle associated with a particular build angle (α) for a circular drill path.

Annular Space - The space surrounding a cylindrical object within a cylinder. The space around a pipe in a borehole is often termed the annulus, and its outer wall may be either the wall of the borehole or the casing.

Barrel - A volumetric unit of measure used in the petroleum industry consisting of 42 gal.

Bentonite - A plastic, colloidal clay, largely made up of the mineral sodium montmorillonite, a hydrated aluminum silicate. For use in drilling fluids, bentonite has a yield in excess of 85 bbl/ton. The generic term "bentonite" is neither an exact mineralogical name, nor is the clay of definite mineralogical composition.

Build Angle (β) - Rate of angular change along a drill path measured in degrees per 100 feet.

Circulation - The movement of drilling fluid from the suction pit through pump, drill pipe, bit, annular space in the hole, and back again to the suction pit. The time involved is usually referred to as circulation time.

Circulation, Loss of (or Lost) - The result of drilling fluid escaping into the formation by way of crevices or porous media.

Circulation Rate - The volume flow rate of the circulating drilling fluid usually expressed in gallons or barrels per minute.

Clay - A plastic, soft, variously colored earth, commonly a hydrous silicate of alumina, formed by the decomposition of feldspar and other aluminum silicates. See also Attapulgite, Bentonite, High Yield, Low Yield, and Natural Clays. Clay minerals are essentially insoluble

in water but disperse under hydration, shearing forces such as grinding, velocity effects, etc., into the extremely small particles varying from submicron to 100-micron sizes.

D - Depth of the horizontal drill path.

Diameter - The distance across a circle measured through the center. In the measurement of pipe diameters, the inside diameter-I.D.- is that of the interior circle, whereas the outside diameter-O.D.- is the diameter of the circle formed by the exterior surface of the pipe.

Dog-Leg - The "elbow" caused by a sharp change of direction in the well bore.

Drilling Mud or Fluid - A circulating fluid used in rotary drilling to perform any or all of various functions required in the drilling operation.

Equivalent Circulating Density - For a circulating fluid, the equivalent circulating density in lb/gal equals the hydrostatic head (psi) plus the total annular pressure drop (psi) divided by the depth (ft) and by 0.052.

Entry Point - Point on the earth's surface where the drill bit initially penetrates.

Filter Cake - Filter cake refers to the layer of concentrated solids from the drilling mud that forms on the walls of the borehole opposite permeable formations. Also called mud cake.

Filter-Cake Texture - The physical properties of a cake as measured by toughness, slickness, and brittleness.

Fluid - A fluid is a substance readily assuming the shape of the container in which it is placed. The term includes both liquids and gases. It is a substance in which the application of every system of stresses (other than hydrostatic pressure) will produce a continuously increasing deformation without any relation between time rate of deformation at any instant and the magnitude of stresses at that instant. Drilling fluids are usually Newtonian and plastic, seldom pseudoplastic, and rarely dilatant fluids.

Fluid Flow - The state of fluid dynamics of a fluid in motion is determined by the type of fluid (e.g., Newtonian, plastic, pseudoplastic, dilatant), the properties of the fluid such as viscosity and density, the geometry of the system, and the velocity. Thus, under a given set of conditions and fluid properties, the fluid flow can be described as plug flow, laminar (called also Newtonian, streamline, parallel, or viscous) flow, or turbulent flow. See above terms and Reynolds number.

Fluid Loss - Measure of the relative amount of fluid lost (filtrate) through permeable formations or membranes when the drilling fluid is subjected to a pressure differential.

H - Horizontal distance from the point of entry to the point the drill bit transverses to the horizontal plane.

Horsepower(HP) - $\dfrac{\text{Force(lb)} \times \text{speed(ft/min)}}{33{,}000}$

The rate of doing work (transferring energy) equivalent to lifting 33,000 lb 1 ft/min (33,000 ft-lb/min). This is also 550 ft-lb/sec.

Hydraulic Horsepower(HHP) - $\dfrac{\text{Circulation rate(GPM)} \times \text{differential Pressure(psi)}}{1{,}714}$

Instaneous Radius of Curvature - The radius of curvature along a spiral drill path measured at a particular point.

Jet Bit - A drilling bit having nozzles through which the drilling fluid is directed in a high velocity stream.

Key Seat - That section of a hole, usually of abnormal deviation and relatively soft formation, which has been eroded or worn by drill pipe to a size smaller than the tool joints or collars. This keyhole type configuration will not allow these members to pass when pulling out of the hole.

Kinematic Viscosity - The kinematic viscosity of a fluid is the ratio of the viscosity (e.g., cp in g/cm-sec) to the density (e.g., g/cc) using consistent units. In several common commercial viscometers the kinematic viscosity is measured in terms of the time of efflux (in seconds) of a fixed volume of liquid through a standard capillary tube or orifice.

Kink Radius of Curvature - The smallest radius of curvature in a undulated section of the drill path.

Laminar Flow - Fluid elements flowing along fixed streamlines which are parallel to the walls of the channel of flow. In laminar flow, the fluid moves in plates or sections with a differential velocity across the front which varies from zero at the wall to a maximum toward the center of flow. Laminar flow is the first stage in a Newtonian fluid; it is the second stage in a Bingham plastic fluid. This type of motion is also called parallel, streamline, or viscous flow.

Mud - A water-or-oil-base drilling fluid whose properties have been altered by solids, commercial and/or native, dissolved and/or suspended. Used for circulating out cuttings and many other functions while drilling a well. Mud is the term most commonly given to drilling fluids.

Mud Pit - Earthen or steel storage facilities for the surface mud system. Mud pits which vary in volume and number are of two types: circulating and reserve. Mud testing and conditioning are normally done in the circulating pit system.

Mud Program - A proposed or followed plan or procedure for the type(s) and properties of drilling fluid(s) used in drilling a well with respect to depth. Some factors that influence the mud program are the casing program and such formation characteristics as type, competence, solubility, temperature, pressure, etc.

Mud Pumps - Pumps at the rig used to circulate drilling fluids.

Newtonian Fluid - The basic and simplest fluids from the standpoint of viscosity consideration in which the shear force is directly proportional to the shear rate. These fluids will immediately begin to move when a pressure or force in excess of zero is applied. Examples of Newtonian fluids are water, diesel oil, and glycerine. The yield point as determined by direct-indicating viscometer is zero.

Pressure-Drop Loss - The pressure lost in a pipeline or annulus due to the velocity of the liquid in the pipeline, the properties of the fluid, the condition of the pipe wall, and the alignment of the pipe. In certain mud-mixing systems, the

loss of head can be substantial.

Pseudoplastic Fluid - A complex non-Newtonian fluid that does not possess thixotropy. A pressure or force in excess of zero will start fluid flow. The apparent viscosity or consistency decreases instantaneously with increasing rate of shear until at a given point the viscosity becomes constant. The yield point as determined by direct-indicating viscometer is positive, the same as in Bingham plastic fluids; however, the true yield point is zero. An example of a pseudoplastic fluid is guar gum in fresh or salt water.

Radius - Radius of curvature of the drill path, $R=l_s/2 \cot(A/_2)$.

Rate of Shear - The rate at which an action, resulting from applied forces, causes or tends to cause two adjacent parts of a body to slide relatively to each other in a direction parallel to their plane of contact. Commonly given in rpm.

Reynolds Number - A dimensionless number. Re, that occurs in the theory of fluid dynamics. The diameter, velocity, density, and viscosity (consistent units) for a fluid flowing through a cylindrical conductor are related as follows:

Re=(diameter)(velocity)(density)/(viscosity)
or
$=DV\rho/\mu$.

The number is important in fluid hydraulics calculations for determining the type of fluid flow, i.e., whether laminar, or turbulent. The transitional range occurs approximately from 2000 to 3000; below 2000 the flow is laminar, above 3000 the flow is turbulent.

Shear Strength - A measure of the shear value of the fluid. The minimum shearing stress that will produce permanent deformation.

Stuck - A condition whereby the drill pipe, casing, or other devices inadvertently become lodged in the hole. May occur while drilling is in progress, while casing is being run in the hole, or while the drill pipe is being hoisted. Frequently a fishing job results.

Tool Joint - A drill-pipe coupler consisting of a pin and a box of various designs and sizes. The internal design of tool joints has an important effect on mud hydrology.

Torque - A measure of the force or effort applied to a shaft causing it to rotate. On a rotary rig this applies especially to the rotation of the drill stem in its action against the bore of the hole. Torque reduction can usually be accomplished by the addition of various drilling-fluid additives.

Tricone Bit - A type of rock bit in which each of three toothed, conical cutters is mounted on friction reducing bearings. The bit body is fitted with nozzles--jets--through which the drilling fluid is discharged.

Velocity - Time rate of motion in a given direction and sense. It is a measure of the fluid flow and may be expressed in terms of linear velocity, mass velocity, volumetric velocity, etc. Velocity is one of the factors which contribute to the carrying capacity of a drilling fluid.

Viscosity - The internal resistance offered by a fluid to flow. This phenomenon is attributable to the attractions between molecules of a liquid, and is a measure of the combined effects of adhesion and cohesion to the effects of suspended particles, and to the liquid environment. The greater this resistance, the greater the viscosity.

Wall Cake - The solid material deposited along the wall of the hole resulting from filtration of the fluid part of the mud into the formation.

Washover Pipe - An accessory used to go over the outside of tubing or drill pipe, thus to clean out the annular space and permit recovery or movement.

Water Table - The underground level at which water is found.

Yield Value - The yield value (commonly called "yield point") is the resistance to initial flow, or represents the stress required to start fluid movement. This resistance is due to electrical charges located on or near the surfaces of the particles. The values of the yield point and thixotropy, respectively, are measurements of the same fluid properties under dynamic and static states. The Bingham yield value, reported in lb/100 sq ft, is determined by the direct-indicating viscometer by subtracting the plastic viscosity from the 300-rpm reading.

α - Build Angle - Rate of angular change along a drill path measured in degrees per 100 feet.

β - Exit Angle - Angle of incline the drill path forms with the horizontal as the drill bit returns to the earth's surface.

Note: Most of the above definitions have been taken from IMCO (1974), The University of Texas at Austin (1974).

APPENDIX B

CALCULATIONS FOR IMPORTANT CONSIDERATIONS IN HORIZONTAL BORING

B.1 INTRODUCTION

This appendix contains the basic soil mechanics, strength of materials, and fluid dynamics calculations behind the analysis of the operational characteristics of the Maneuverable Penetration System(MPS) in soft ground as defined in Table B.1. The underlying assumptions for these calculations have been made from an intuitive standpoint of what might be happening in the drill hole and do not reflect the results of laboratory tests or detailed field data comparisons.

These assumptions are as follows:

1) The clay soil is assumed to be in an undrained condition since 99% of the bore hole is located below the water table. Clay strengths are given in Table B.1, repeated here for convenience. Any sand encountered will be completely saturated, and total stresses are equal to effective stresses.

2) The maximum depth below the ground surface is 500 ft(153 m) and the optimal horizontal drill hole distance is 5000 ft(1525 m).

Table B.1 Shear Strength of Cohesive Soils
(Terzaghi and Peck, 1967)

<table>
<tr><th>$S_u = \frac{1}{2} q_u$ (tsf)</th><th>Consistency</th><th>Unit Weight (pcf)</th></tr>
<tr><td>0-0.125</td><td>Very Soft</td><td rowspan="2">100-200</td></tr>
<tr><td>0.125-0.25</td><td>Soft</td></tr>
<tr><td>0.25-0.50</td><td>Medium</td><td>110-130</td></tr>
<tr><td>0.50-1.0</td><td>Stiff</td><td rowspan="2">120-140</td></tr>
<tr><td>1.0-2.0</td><td>Very Stiff</td></tr>
<tr><td>2.0</td><td>Hard</td><td>130+</td></tr>
</table>

3) Any estimations of the maximum equipment limits for bending stresses have not considered continuous, long-term operating conditions under these bending stresses and how this will affect the future performance of the equipment.

B.2 MINIMUM S_u FOR THRUST APPLICATOR OPERATION

First, the capabilities of the DRILCO thrust applicator will be calculated with a modified version which has the following characteristics:

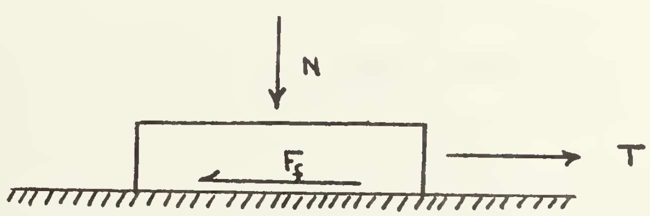
Thrust Applicator O.D. - 8 in

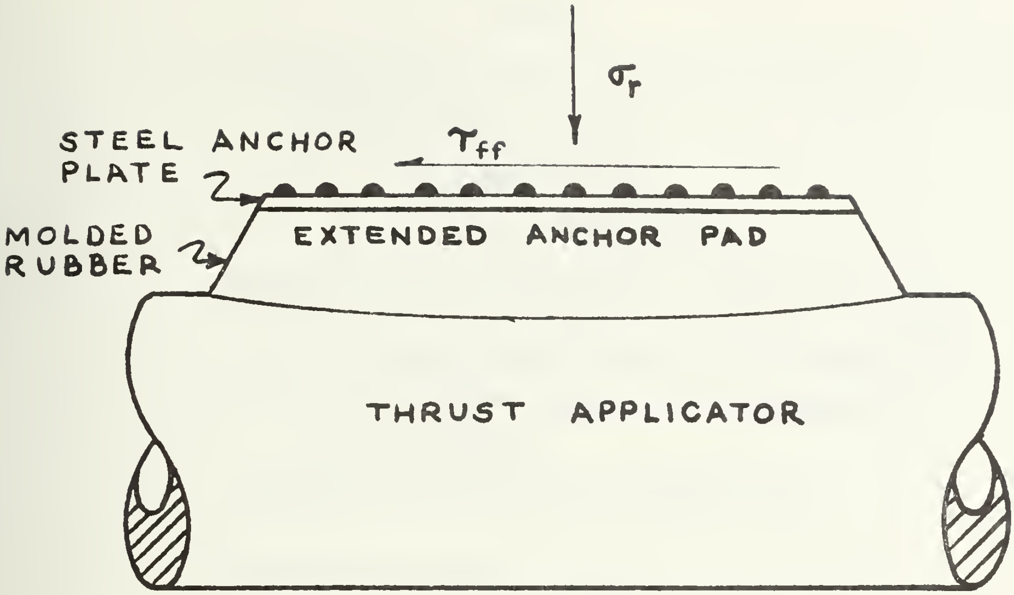
Anchor pads

- 1 set of piston pads (3 pads/set)
- 3 sets of cylinder pads
- Pad dimensions - 1-1/2 x 8 in
- Single pad area - 12 in^2
- Total pad area available - 144 in^2

The undrained shear strength for clay and the shear strength for sand are calculated using equations shown in Figure B.1. To calculate the minimum S_u required for the thruster to pull its hose down the hole, the frictional force on the hose must first be calculated.

Figure B.2 Friction Forces Acting on Thruster Hose





NOT TO SCALE:

Cohesionless soil:

$$\tau_{ff} = S_s = S_d = \bar{\sigma}_r \tan \phi$$

Cohesive soil:

$$S_s \simeq C = S_u$$

$$S_s \simeq S_u \simeq F_s/A_t$$

F_s = Shearing Force

A_t = Total Pad Area

$\bar{\sigma}_r$ = Anchoring stress applied across anchor pad surface area

FIGURE B.1 Shear Strength Formulas

To overcome static friction $T > F_f = \mu N$

for sands $\mu = \tan \bar{\phi}$

assume $\bar{\phi} = 35^o$

$\mu = \tan 35^o = 0.7$

Weight of the individual thruster hoses:

1-1" drilling fluid hose - @ 24.4#/100'

2-1/2" hydraulic hose - @ 13.4#/100'

N=51.2#/100'

$F_f = (0.7)(51.2) = 35.84 \simeq 36$#/100'

For 1000' tunnel length

$F_f = 360$#

Then, S_u is determined for a thrust applicator as follows:

Thrust Applicator - 8" O.D.

12 pads - (1.5" x 8")

$A_t = 144$ sq in

$S_s = \frac{\sigma_r}{A_t} \tan\bar{\phi} = F_f/A_t = 360/144 = 2.5$ psi

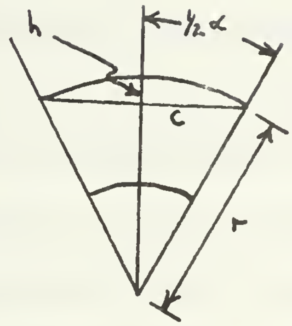
For sands:

$S_s \simeq S_u = 0.18$ tsf

B.3 BEARING CAPACITY AND CONTRACT STRESS CALCULATIONS

Before any bearing capacity or contact stress calculations are performed, the size of the anchor pad must be redesigned for soft ground conditions for the dimensions stated in Section B.2. An estimation of a redesigned thruster pad is based on finding the length of a chord for the same degree of arc at a

larger radius.



$$c = 2\sqrt{h(d-h)}$$

$$h = r(1-\cos\alpha)$$

where c = cord length

h = cord diameter

$$\frac{C_2^2}{C_1^2} = \frac{r_2 d_2}{r_1 d_1}$$

$$C_2 = C_1 \sqrt{\frac{r_2 d_2}{r_1 d_1}}$$

if, $C_1 = 1.06$ in

$d_1 = 5.75$ in

$d_2 = 8.0$ in

then, $C_2 = 1.47$ in ≈ 1.5 in (3.8 cm)

<u>Calculation of Maximum Contact Stress</u> The maximum contact stress is based on the maximum allowable hydraulic pressure the thrust applicator is able to apply to the anchor shoe diaphragm before rupture occurs. For the particular calculation this value of maximum pressure will be 500 psi(3450 kN/m^2), however, this is a measured value for the 5-3/4 in(14.6 cm) O.D. thrust applicator.

For a thrust applicator anchor pad:

$$A_H = 1.06 \times 6 = 6.36 \text{ in}^2$$

$$A_c = 1.5 \times 8 = 12 \text{ in}^2$$

$$\Delta P_H = 500 \text{ psi}$$

$$\sigma_{c_{max}} = 500\left(\frac{6.36}{12}\right) = 265 \text{ psi}$$

$$\sigma_{c_{max}} = 19.08 \text{ tsf} = 1826 \text{ kN/m}^2$$

where: A_H=anchor pad area in contact with hydraulic fluid

A_C=anchor pad area in contact with the soil

For the CONOCO deflection shoe:

ΔP_H=600 psi is the constant pressure applied to the drive piston (2 in dia.) for extending the deflection shoe

$$A_H = \frac{\pi d^2}{4} = \frac{\pi (2)^2}{4} = 3.142 \text{ in}^2$$

$$A_c = 4 \times 8 = 32 \text{ in}^2 \text{ (estimated dimensions)}$$

$$\Delta P_H = 600 \text{ psi}$$

$$\sigma_{c_{max}} = 600\left(\frac{3.142}{32}\right) = 58.9 \text{ psi}$$

$$\sigma_{c_{max}} = 4.24 \text{ tsf} = 406 \text{ kN/m}^2$$

Minimum Contact Stress The minimum contact stress is defined as the pressure required to first move the anchor pad. This pressure for the 5-3/4 in(14.6 cm) O.D. thrust applicator is 100 psi(690 kN/m^2).

$$A_H = 6.36 \text{ in}^2$$

$$A_c = 12 \text{ in}^2$$

$$\Delta P_H = 100 \text{ psi}$$

$$\sigma_{c_{max}} = 100\left(\frac{6.36}{12}\right) = 53 \text{ psi}$$

$$\sigma_{c_{max}} = 3.6 \text{ tsf} = 364 \text{ kN/m}^2$$

Bearing Capacity Calculations The bearing capacity has been calculated for the type of failure shown in Figure 3.8.

In clay: $q = N_c\ S_u + P_A$

where N_c=bearing capacity factor
P_A=annular pressure

N_c (Rectangular shape)=(0.84+0.16 B/L)N_c (Square)

N_c (Square)=6.2

Table B.2 Annular Pressure for 2-3/8 in O.D. Dyna-Drill with 2-3/8 in Dia. Drill Pipe and a 4-1/2 in Hole

Drill Hole Length - ft	P_A - tsf
1000	1.58
2000	3.17
3000	4.75
4000	6.34
5000	7.92

Thrust Applicator (pad dimensions 1-1/2 x 8 in)

$$N_c = \left[0.84 + \left(\frac{1.5}{8}\right)(0.16)\right] 6.2 = 5.39$$

$$S_u = 0.25 \text{ tsf}$$

$$L = 1000 \text{ ft}$$

$$q_{ult} = 5.39(0.25) + 1.58 = 2.93 \text{ tsf}$$

Deflection Shoe (dimensions 4 x 6 in)

$$N_c = \left[0.84 + \left(\frac{4}{6}\right) 0.16\right] 6.2 = 5.87$$

$$q_{ult} = 5.87(0.25) + 1.58 = 12.36 \text{ tsf}$$

In Sand:

$$q_{ult} = \tfrac{1}{2} S_\gamma \gamma B N_\gamma + P_a$$

$$S_\gamma = (1.0 - 0.4 B/L) \qquad \text{(Vesic, 1973)}$$

γ = unit weight of bearing soil

Soil Parameters: $\bar{\phi} = 30^\circ$

$\gamma = 110$ pcf

$\gamma_b = 47.6$ pcf

$N_\gamma = 22.5$ (loose sand)

Thrust Applicator:

$$S_\gamma = 1 - 0.4\left(\frac{1\tfrac{1}{2}}{8}\right) = 0.925$$

For $L = 5000$ ft

$$q_{ult} = 0.925(0.5)(47.6)(0.125)(22.5) + 15{,}840$$

$$q_{ult} = 7.96 \text{ tsf}$$

Deflection Shoe:

$$S_\gamma = 1 - 0.4\left(\frac{4}{6}\right) = 0.733$$

For $L = 5000$ ft

$$q_{ult} = 0.733(0.5)(47.6)(0.33)(22.5) + 15,840$$

$$q_{ult} = 8.0 \text{ tsf}$$

B.4 MAXIMUM EXIT ANGLE

The maximum exit angle calculations have been performed for both sand and clay. In order to include all the forces acting on the MPS, a few of the drilling fluid flow calculations have been performed in this section.

Fluid Flow Calculations The Rheoplot data needed to calculate the Reynold's number for the pseudoplastic drilling mud has kindly been provided by Mr. M. Lowrance from Milchem, Drilling Fluids Division.

For these following calculations the generalized Reynold's number was adopted. The pseudoplastic stress-deformation relationship was modeled using the familiar Power Law $\tau = K(du/dy)^n$.

Generalized Reynold's Number:

$$N_R' = \frac{D^n V^{2-n} \rho}{K^*}$$

The empirical corrected K*:

$$K^* = K(8^{n-1})\left(\frac{3n+1}{4n}\right)^n$$

n is the slope of the straight line on the Rheoplot

When this relationship is plotted on log-log paper it takes the form:

$$\log \tau = \log K + n \log(du/dy)$$

where the log K is the τ intercept at n log(du/dy)=1 and n is the slope of the straight line as shown in Figure B.3. Another way to look at the log-log plot of this relationship is as a stress-strain diagram for the mud slurry and the log K value is the dynamic yield point of the mud slurry and the "n" value is the dynamic viscousity of the fluid.

From Figure B.3 the following values were estimated:

$$K = 2 \times 10^{-2} \text{ lbs-sec/ft}^2$$

$$n = 0.404$$

Therefore,

$$K^* = (2 \times 10^{-2})(8^{-.592})(1.37)^{0.404}$$

$$K^* = 6.63 \times 10^{-3} \text{ lb-sec/ft}^2$$

The drag forces of fluid flowing past a neutrally buoyant drill pipe are calculated with a 2-3/8 in(6.0 cm) diameter steel drill pipe in a 4-1/2 in(11.4 cm) hole.

$$V = Q/A$$

$$Q = \frac{25}{7.48(60)} = 0.0557 \text{ ft}^3/\text{sec}$$

$$A = \frac{\pi(d_1^2 - d_2^2)}{4} = 0.0797 \text{ ft}^2$$

$$V = 0.7 \text{ ft/sec}$$

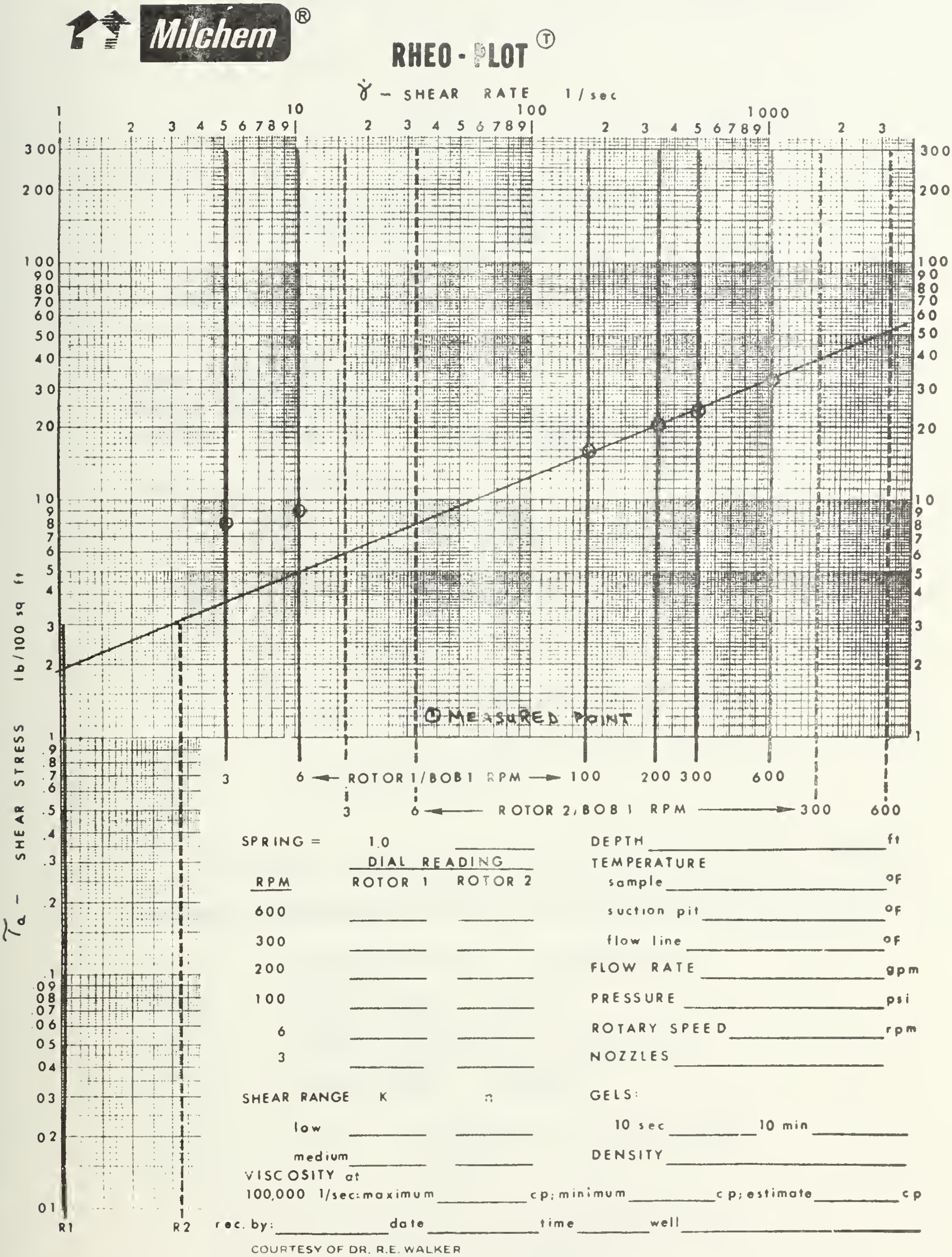


FIGURE B.3 Milchem Rheoplot[R]

The mud density for a 21 lb/bbl drilling mud is:

$$\rho = 65.834 \text{ lbs/ft}^3$$

$$d = \frac{d_1^2 - d_2^2}{d_1 + d_2} = 2.125 \text{ in} = 0.177 \text{ ft}$$

$$N_R' = \frac{(0.177)^{0.404}(0.7)^{1.57}(65.824)}{6.63 \times 10^{-3}\ (32.2)}$$

$$N_R' = 87.8$$

Then the empirical value for CD for laminar flow is:

$$C_D = \frac{1.28}{\sqrt{N_R'}} = \frac{1.28}{\sqrt{87.8}} = 0.137$$

The drag force is:

$$D = 1/2\ \rho\ C_D V^2 S$$

where S = Surface area / linear foot

$$= \frac{\pi d (1)}{LF} = 0.62 \text{ ft}^2/\text{LF}$$

$$D = \frac{1/2\ (65.824)(0.1097)(0.49)(0.62)}{32.2}$$

$$D = 0.034 \text{ lbf/LF}$$

Similar values are calculated for the DRILCO thrust applicator with the following dimensions:

Thruster O.D. - 5-3/4 in

Hole Size - 7 in

Cable Diameter - 1-1/2 in

Flow Rate - 25 GPM

$$V = Q/A = \frac{0.0557}{0.098} = 0.568 \text{ ft/sec}$$

$$N_R' = \frac{(0.076)^{0.404}(0.218)^{1.57}(65.824)}{6.63 \times 10^{-3}\ (32.2)} = 10.1$$

and $C_D=0.403$

then the drag force is:

$D=7.7$ lb/100 L.F.

Forces Acting on the Mandrel MPS in Each Section

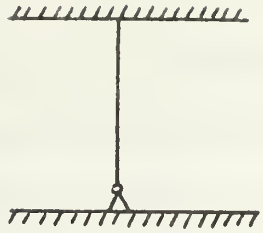
The weight calculations for the mandrel MPS are as follows:

2-3/8" Dyna-Drill	- 60 lbs
4-1/2" bit	- 3 lbs
Navigation equip.	- 15 lbs
Sensing equip.	- 20 lbs
2-3/8" Drill Pipe for 250'	-958 lbs
Total	1056 lbs

The buoyant weight of the drill pipe in a 21 lb/bbl drilling mud was calculated to be 3.83 lbs/L.F.

The following calculations have been made for a horizontal distance of 3000 and 5000 feet in Section II, as described in Chapter 3.

An estimate of the maximum normal force that can be applied to the mandrel MPS system is based on Euler's slender buckling criteria as applied to the drill pipe. The model used for these calculations, as shown below, simulates the condition illustrated by Figure 3.12b for Titan Contractors' drilling rig.



$2\,3/8$ in O.D (1.995 in I.D.) Drill Pipe

$n = 2$

$\ell = 25$ ft

$E = 29{,}000$ ksi

$$P_{crit} = \frac{n \pi^2 E A}{(\ell/r)^2}$$

where $A = \dfrac{\pi(d_1^2 - d_2^2)}{4} = 1.304 \text{ in}^2$

$$r = \sqrt{r_1^2 + r_2^2}/2 = 0.775 \text{ in}^2$$

$$P_{crit} = \frac{2\pi^2(29{,}000)(1.304)}{(387)^2}$$

$$P_{crit} = 4.93 \text{ kips}$$

For the normal force calculations a reduction factor of 1.25 is applied to P_{crit} which gives the maximum allowable normal force to be applied to the drill pipe at the surface (i.e. $F_N = P_{crit}/1.25 = 3.95$ kips)

Table B.3 Forces Acting on the Mandrel MPS in Sand

<table>
<tr><td>Section</td><td>I</td><td colspan="2">II</td><td>III</td></tr>
<tr><td>Normal Force F_N(lbf)</td><td colspan="4">3950</td></tr>
<tr><td>Frictional Force F_f(lbf)</td><td>469</td><td colspan="2">47</td><td>—</td></tr>
<tr><td rowspan="2">Drag Force D(lbf)</td><td rowspan="2">25</td><td>3000</td><td>5000</td><td rowspan="2">25</td></tr>
<tr><td>128</td><td>212</td></tr>
</table>

The friction factor for Section II takes into account the friction encountered at the two bends in the drill path as shown in Figure 3.14 and explained in Chapter 3. For a horizontal distance equal to 3000 ft (915 m)

$$F=D+1.1F_f=176.5+516=693$$

$$W=1058 \text{ lbs}$$

$$F_N=3950$$

The exit angle criteria as defined in Section 3.7,

$$(F_N-F)/W = \frac{3950-693}{1058} = 3.08$$

For a horizontal distance of 5000 ft(1525 m)

$$(F_N-F)/W=3.05$$

Therefore, if the drill pipe remains neutrally buoyant in Section II (i.e. zero soil friction) then the mandrel MPS can drill out of the hole along an exit angle of 90^o.

Maximum Horizontal Distance for Mandrel MPS These calculations take into consideration the effects of soil resistance along Section II and how this will change the maximum horizontal distance the mandrel MPS can drill before it has zero normal force at the bit. The maximum thrust available for the horizontal section is the normal force plus the angular weight component minus the friction forces along Section I (T=3950+884-512=4322 lbf). To find the maximum horizontal distance the MPS can travel, divide the

available thrust by the combined linear footage values for friction and drag forces.

$$T = D + F_f$$

where $D = 0.0425$ lbf/L.F.

$F = 0.7(3.83$ lbs/L.F.)

Note: The buoyant weight of the drill pipe is 3.83 lbs/L.F.

$$H_{max} = \frac{4322 \text{ lbs}}{2.72 \text{ lbs/L.F.}} = 1589 \text{ L.F.}$$

Forces Acting on Thrust Applicator MPS in Sand

The weight estimations for the thrust applicator MPS are as follows:

Thrust Applicator	-	100 lbs
Hydraulic Motor	-	10 lbs
7" bit	-	6 lbs
Deflection Shoe	-	10 lbs
Orientation Motor	-	10 lbs
Hose @ 51.2#/100'	-	128 lbs
Total	-	279 lbs

To calculate what the maximum pulling capacity available(MPCA) for the thrust applicator would be, the Mohr-Coulomb failure criteria was adapted in the following manner:

$$\tau_f = \bar{\sigma}_n \tan \bar{\phi}$$

Assuming K=1 (coefficient of earth pressure) at depth equal 250 ft then $\bar{\sigma}_n$ is the average normal stress required for anchoring. A worst condition case will

be analyzed:

loose sand, $\bar{\phi}=30^\circ$, $\gamma_t=110$ pcf

$\bar{\sigma}_h = z\gamma_b = 11{,}900 \text{ lbs/ft}^2 \ (570 \text{ kN/m}^2)$

$\tau_f = 11{,}900 \tan 30^\circ = 6871 \text{ lbs/ft}^2 \ (329 \text{ kN/m}^2)$

$MPCA = \tau_f A_t$

where A_t=total anchor pad contact area

The redesigned, enlarged anchor pads with dimensions 1-1/2 x 8 in(3.8 x 20.3 cm) are combined in three sets of cylinder anchors for the following calculations.

Table B.4 Forces Acting on the Thrust Applicator MPS in Sand

Section	I	II	III
MPCA (lbf)	5153		
Frictional Force F_f(lbf)	98	9.8	—
Drag Force D(lbf)	4	$\frac{3000}{23}$ $\frac{5000}{39}$	4

For a horizontal distance of 3000 ft(915 m)

$$\frac{MPCA-F}{W}=\frac{5153-1140}{279}=14.4$$

where now $F=D+1.1F_f+T$

T is assumed to be equal to 1000 lbf

For a horizontal distance of 5000 ft(1525 m)

$$\frac{MPCA-F}{W} = 14.3$$

Now, if the worst condition is assumed where the thruster hose is dragging on the bottom of the hole in Section II, how will the thruster MPS perform? The friction force for Section II is F_f'=1075 lbs (@ H=3000 ft); therefore,

$$F=D+1.1F_f+F_f'+T=2249 \text{ lbs}$$

For a horizontal distance of 3000 ft(915 m)

$$\frac{MPCA-F}{W}=\frac{5153-2249}{534}=5.4$$

For a horizontal distance of 5000 ft(1525 m)

$$\frac{MPCA-F}{W}=4.2$$

Both of these values indicate the possibility exists for the thrust applicator MPS to exit at an angle up to 90°, even if it must drag its hose behind it in loose sand.

Mandrel MPS Operating in Clay The same mandrel MPS previously mentioned will be used for the following calculations.

Table B.5 Forces Acting on the Mandrel MPS in Clay

Section		I	II		III
Normal Force	F_N (lbf)	3950			
Friction Force in Sticky Clay	F_f (lbf)	2209	221		2209
Friction Force in over-consolidated clay	F_f (lbf)	469	47		469
Drag Force	D (lbf)	25	3000 128	5000ft 212	25

The friction force is different for a sticky clay than an overconsolidated clay as explained in Section 3.9. In addition, the frictional forces are now assumed values based on empirical information referenced in the same section.

For a horizontal distance equal to 3000 ft(915 m)

$$\frac{F_N - F}{W} = \frac{3950-1162}{1056} = 2.64$$

and for a 5000 ft(1525 m) distance,

$$F_N - F_W = 2.56$$

The ratio $F_N - F_W$ is equal to $\sin\beta$ in clay, therefore both distances can be drilled and the maximum angle is 90^o.

The maximum horizontal distance the mandrel can travel along the horizontal section of the drill path was calculated using the previously mentioned relationships. The available thrust(T), after the first bend is as follows:

$$T=F_N+W'-F_f-D=3950+916-2209-25$$

$$T=2632 \text{ lbs}$$

$$H=\frac{T}{F_f}=\frac{2632 \text{ lbs}}{3.83 \text{ lbs/L.F.}}=687 \text{ L.F.}$$

<u>Thrust Applicator MPS Operating in Clay</u> The following calculations are based on the assumption that the only clay the thrust applicator MPS can operate in is overconsolidated, stiff clay (S_u=2.0 tsf).

Table B.6 Forces Acting on the Thrust Applicator MPS in Overconsolidated, Stiff Clay

<table>
<tr><td>Section</td><td>I</td><td colspan="2">II</td><td>III</td></tr>
<tr><td>MPCA (lbf)</td><td colspan="4">3000</td></tr>
<tr><td>Friction Force F_f (lbf)</td><td>329</td><td colspan="2">33</td><td>329</td></tr>
<tr><td rowspan="2">Drag Force D (lbf)</td><td rowspan="2">4</td><td>3000</td><td>5000 ft</td><td rowspan="2">4</td></tr>
<tr><td>23</td><td>39</td></tr>
</table>

For the horizontal distance of 3000 ft(915 m),

$$\frac{MPCA-F}{W}=\frac{3000-722}{279}=8.1$$

Since the thrust applicator system is lighter in weight than the mandrel MPS, the thrust applicator

will surely operate over a distance of 5000 ft, even if it must drag its cables behind it. This, of course, assumes that the hole remains open in the sticky clay environment.

B.5 PRESSURE LOSS CALCULATIONS FOR A MANDREL MPS

The pressure loss throughout the entire mandrel system is of concern when ordering the proper size surface mud pump and when analyzing whether or not the system selected will hydraulically fracture the soil at the drill bit. For these calculations, basic fluid dynamic relationships have been applied.
The mandrel MPS is a 2-3/8 in(6 cm) O.D. Dyna-Drill system.

The pressure along a horizontal pipe was calculated using the Darcy-Weisbach equation:

$$h_f = f\,\frac{L}{d}\,\frac{V^2}{2g}$$

where $d = \dfrac{d_1^{\,2} - d_2^{\,2}}{d_1 + d_2}$

$f = \dfrac{64}{N_R}$ an empirical friction factor for laminar flow

and $\Delta P = \rho_f h_f$

ρ_f = density of the drilling fluid

$$N_R' = 537$$

$$f = \frac{64}{537} = 0.119$$

$$h_f = 0.119\left(\frac{1000}{0.166}\right)\frac{(2.26)^2}{64.4} = 56.86 \text{ ft}/1000 \text{ LF}$$

$$\Delta P = \rho h_f = \frac{65.824(56.86)}{144} = 26 \text{ lbs/in}^2$$

The annular pressure loss for a 4-1/2 in(11.4 cm) diameter hole with a 2-3/8 in(6 cm) diameter drill pipe follows:

$$N_R' = 87.8$$

$$f = 0.729$$

$$d = \frac{d_1^2 - d_2^2}{d_1 + d_2} = 0.177 \text{ ft}$$

$$V = 0.7 \text{ ft/sec}$$

$$h_f = 0.729\left(\frac{1000}{0.177}\right)\left(\frac{0.49}{64.4}\right) = 31.3 \text{ ft}/1000 \text{ LF}$$

$$\Delta P_a = \rho h_f = \frac{31.3(65.824)}{144} = 14.3 \text{ psi}/1000 \text{ LF}$$

ECD Calculations for the Mandrel MPS For the above described mandrel MPS the following ECD calculations were made. ECD or equivalent circulating density is thoroughly explained in Chapter 3. Because of the uncertainty of the exact increase in drilling mud density due to drilling fines, the previously calculated annular pressure is increased by a factor of 1.5. In addition the pressure differential for the depth of the drill bit is added to the annular pressure loss for a horizontal pipe.

For a drill hole at a depth of 100 ft(31 m) and a horizontal distance of 1000 ft(310 m),

$$ECD = \rho + \frac{\Delta P_a}{0.52\ L}$$

$$\rho = 8.83 \text{ lb/gal} \quad \text{for 21 lb/bbl drilling mud}$$

$$\Delta P_a = \frac{100(65.824)}{144} + 21.5 = 67.2 \text{ psi}$$

$$ECD = 8.83 + \frac{67.2}{0.052(1100)} = 10.01 \text{ lbs/gal } (1.21 \text{ g/cm}^3)$$

At a depth of 500 ft(153 m),

$$\Delta P = \frac{500(65.824)}{144} + 21.5 = 250 \text{ psi}$$

$$ECD = 12.04 \text{ lb/gal } (1.46 \text{ g/cm}^3)$$

B.6 PRESSURE LOSS CALCULATIONS FOR A THRUST APPLICATOR MPS

The pressure loss calculations for the thrust applicator apply the same basic relationships already mentioned. The thrust applicator is 5-3/4 in(14.6 cm) in diameter with a 1-1/2 in(3.8 cm) O.D. trailing hose operating in a 7 in(17.7 cm) diameter hole.

The pressure loss within the drilling slurry hose is as follows:

$$V = Q/A = 10.13 \text{ ft/sec}$$

$$N_R' = 3.76 \times 10^3$$

$$f = 64/N_R' = 0.017$$

$$h_f = 0.017\left(\frac{1000}{0.083}\right)\frac{(10.13)^2}{64.4} = 326 \text{ ft/LF}$$

$$\Delta P = \rho h_f = 149 \text{ psi/1000 LF}$$

Annular Pressure Loss

$$d = \frac{d_1^2 - d_2^2}{d_1 + d_2} = \frac{49 - 2.25}{51.25} = 0.912 \text{ in} = 0.076 \text{ ft}$$

$$N_R' = \frac{(0.076)^{0.404}(0.218)^{1.59}(65.824)}{6.63 \times 10^{-3}\ (32.2)} = 10.1$$

$$f = \frac{64}{10.1} = 6.34$$

$$h_f = 6.34\left(\frac{1000}{0.076}\right)\frac{(0.218)^2}{64.4} = 61.56 \text{ ft}/1000 \text{ LF}$$

$$\Delta P = h_f\, \rho = \frac{(61.56)(65.824)}{144} = 28.1 \text{ psi}/1000 \text{ LF}$$

ECD Calculations At 100 ft(31 m) deep and at a horizontal distance of 1000 ft(310 m),

$$ECD = 8.83 + \frac{73.81}{0.052(1100)} = 10.12 \text{ lb/gal}(1.22 \text{ g/cm}^3)$$

and at 500 ft(153 m) deep

$$ECD = 12.3 \text{ lb/gal}(1.49 \text{ g/cm}^3)$$

APPENDIX C

SPECIFICATIONS ON DOWNHOLE MOTORS

C.1 INTRODUCTION

Four downhole motors are discussed in this appendix because of their previous application to directional drilling in general, or specifically with horizontal directional drilling. These four motors are: Dyna-Drill, Turbo-drill, the electric motor, and the hydraulic motor. Presented in this order, the various drawings will provide the necessary level of understanding for this thesis. If the reader desires more detail, he is encouraged to contact the individuals at the specific company on the List of Contributors.

C.2 DYNA-DRILL

The Dyna-Drill was born as a result of an idea Mr. Wallace Clark had when he saw a Moyno[R] pump operating as an auxillary piece of equipment on an oil rig. The rotor and stator for the Dyna-Drill shown in Figure C.1 are illustrated in Figure C.2a while the basic principle of operation of the Moyno

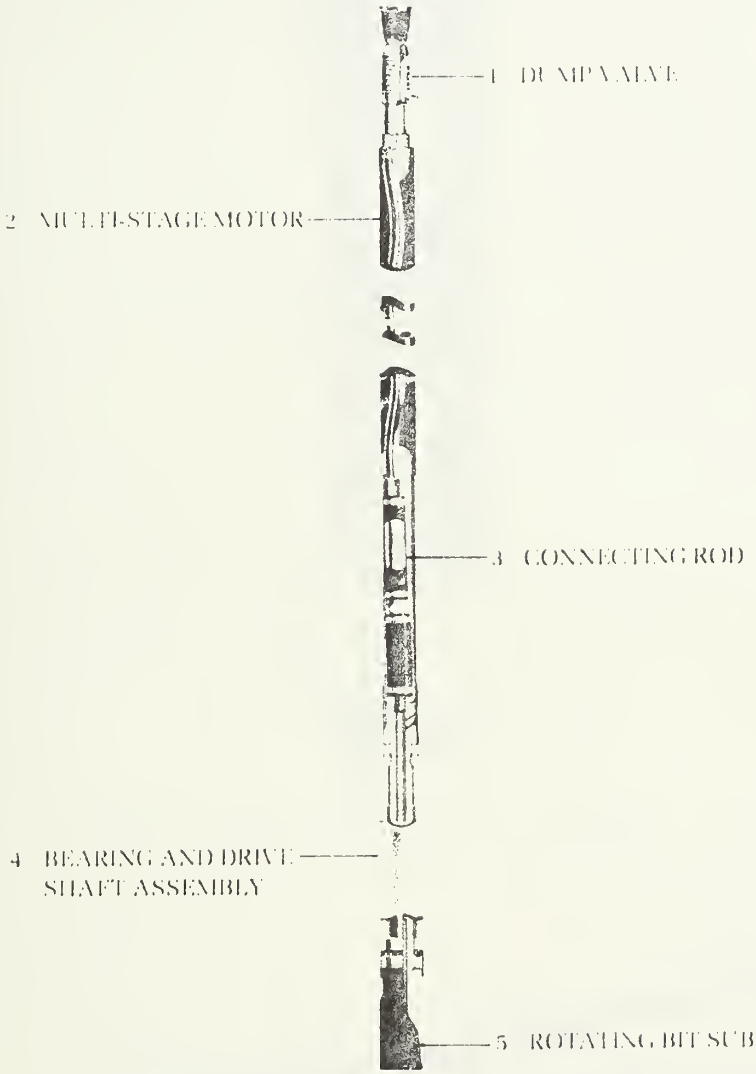


FIGURE C.1 Dyna-Drill (After Dyna-Drill, 1975)

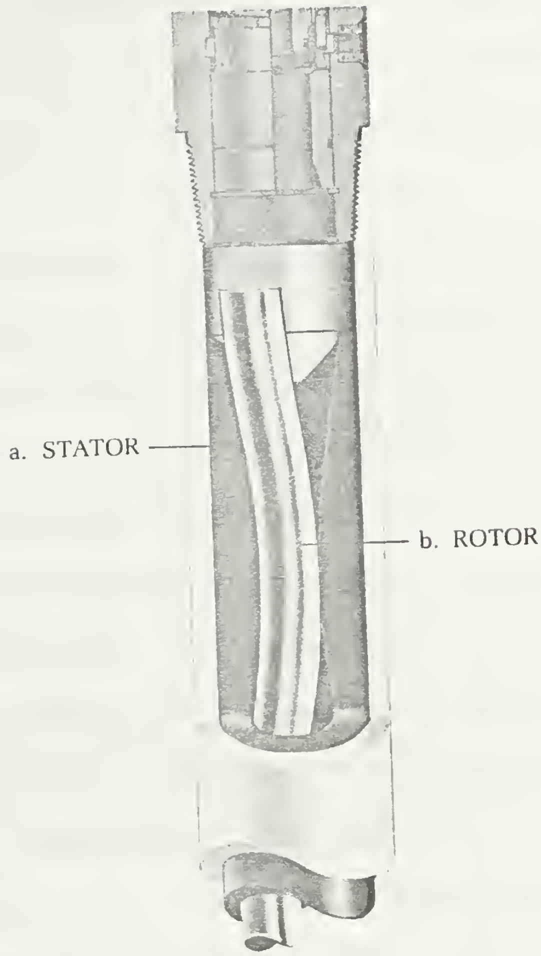


FIGURE C.2(a) Dyna-Drill Stator
(After Dyna-Drill,1975)

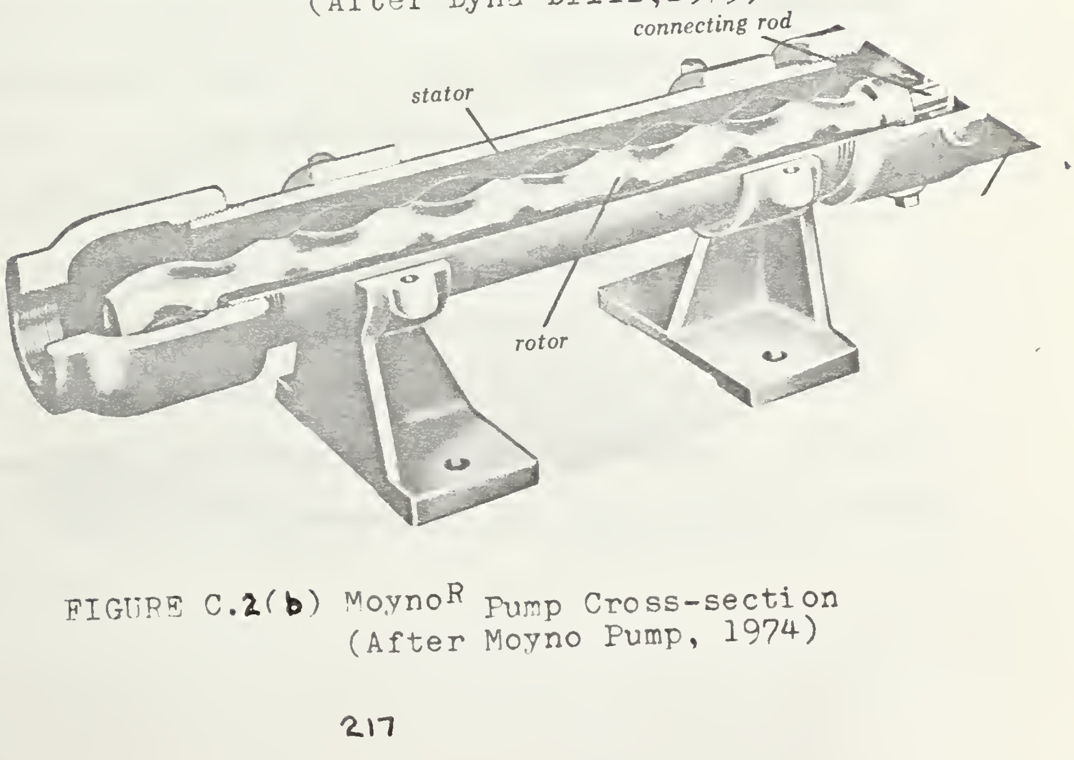


FIGURE C.2(b) Moyno^R Pump Cross-section
(After Moyno Pump, 1974)

pump is illustrated in Figure C.2b. Smith International, Incorporated supported the initial development of this downhole motor, thereby forming the Dyna-Drill Company in 1964.

The Dyna-Drill is essentially a multi-stage Moyno pump operating in reverse as a motor which comprises about one-half of the length of the tool. The obround, spiral stator shown in Figure C.3 is made from synthetic rubber which is compressively fit to reduce slippage. The stator houses the solid steel rotor which has a regular sinusoidal longitudinal wave pattern shape which moves eccentrically within the stator while rotating. The upper end of this rotor is free ended while the bottom end is attached to the connecting rod. The connecting rod in Figure C.4 consists of a universal joint that converts the eccentric motion of the rotor to concentric motion required for the drive shaft. The bit sub in Figure C.5 is at the bottom end of the drive shaft and is the only external moving part.

As the drilling fluid is pumped downward between the stator and the rotor, the rotor is displaced and rotated within the stator as the fluid flows along the spiral path of the stator. This in turn powers the connecting rod, drive shaft, bit sub and bit.

FIGURE C.3 End View of Dyna-Drill Stator
(After Dyna-Drill, 1975)

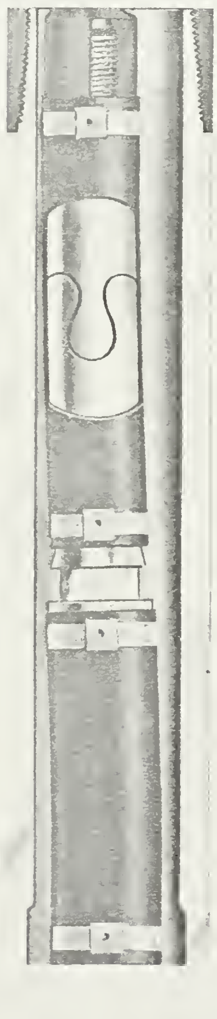


FIGURE C.4 Dyna-Drill Connecting Rod

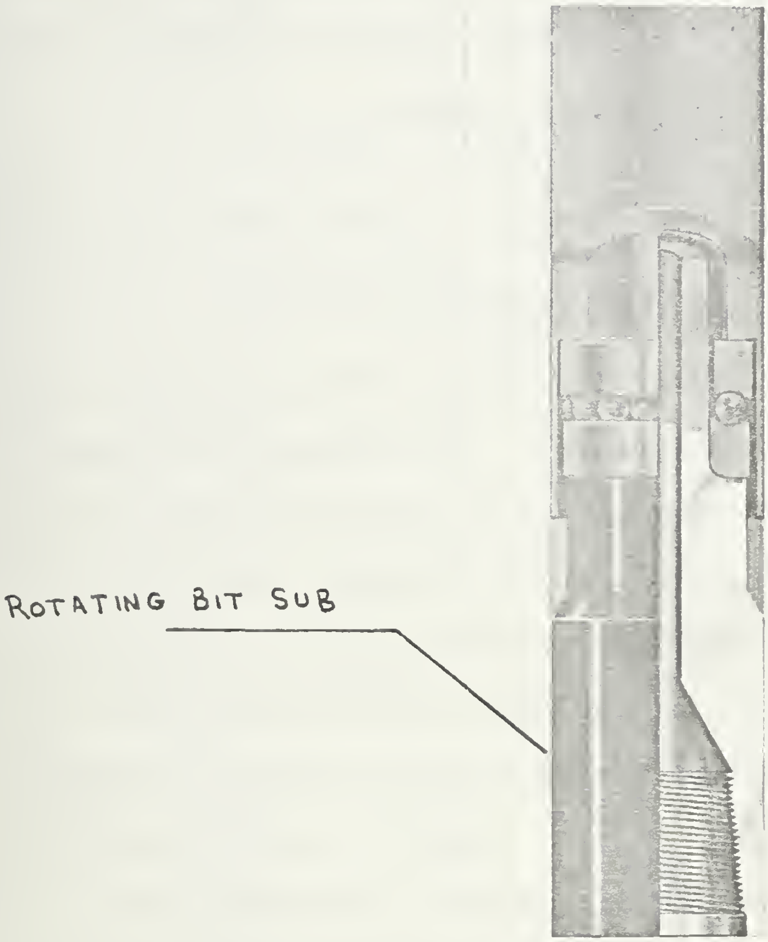


FIGURE C.5 Bit Sub

Since the motor is a positive displacement motor, the hydraulic horsepower and torque output are a function of the pressure loss across the motor.

$$H.P._{Hyd} = \frac{GPM(\Delta Pressure)}{1714}$$

$$Efficiency = \eta = \frac{H.P._{mech}}{H.P._{Hyd}}$$

$$Torque = \frac{H.P._{mech}(63{,}025)}{RPM} \quad (in\text{-}lb)$$

The operational RPM can also be estimated from the change in pressure (ΔP). Therefore, the pressure loss across the motor is a very important factor when drilling with a Dyna-Drill. If the ΔP increases rapidly to a reading greater than twice the operating ΔP, the Dyna-Drill has stalled. The bit weight should be removed quickly when a Dyna-Drill stalls to prevent extensive damage.

The Dyna-Drill configuration lends itself either to a straight housing or bent housing assembly as shown in Figure C.6. The straight housing assembly is used with a bent or articulated sub while the bent housing is attached to the drill pipe with a standard connection. The bent housing tool has a set of ribs on the underside of the tool directly above the bend to help prevent the drill bit from dropping down unintentionally.

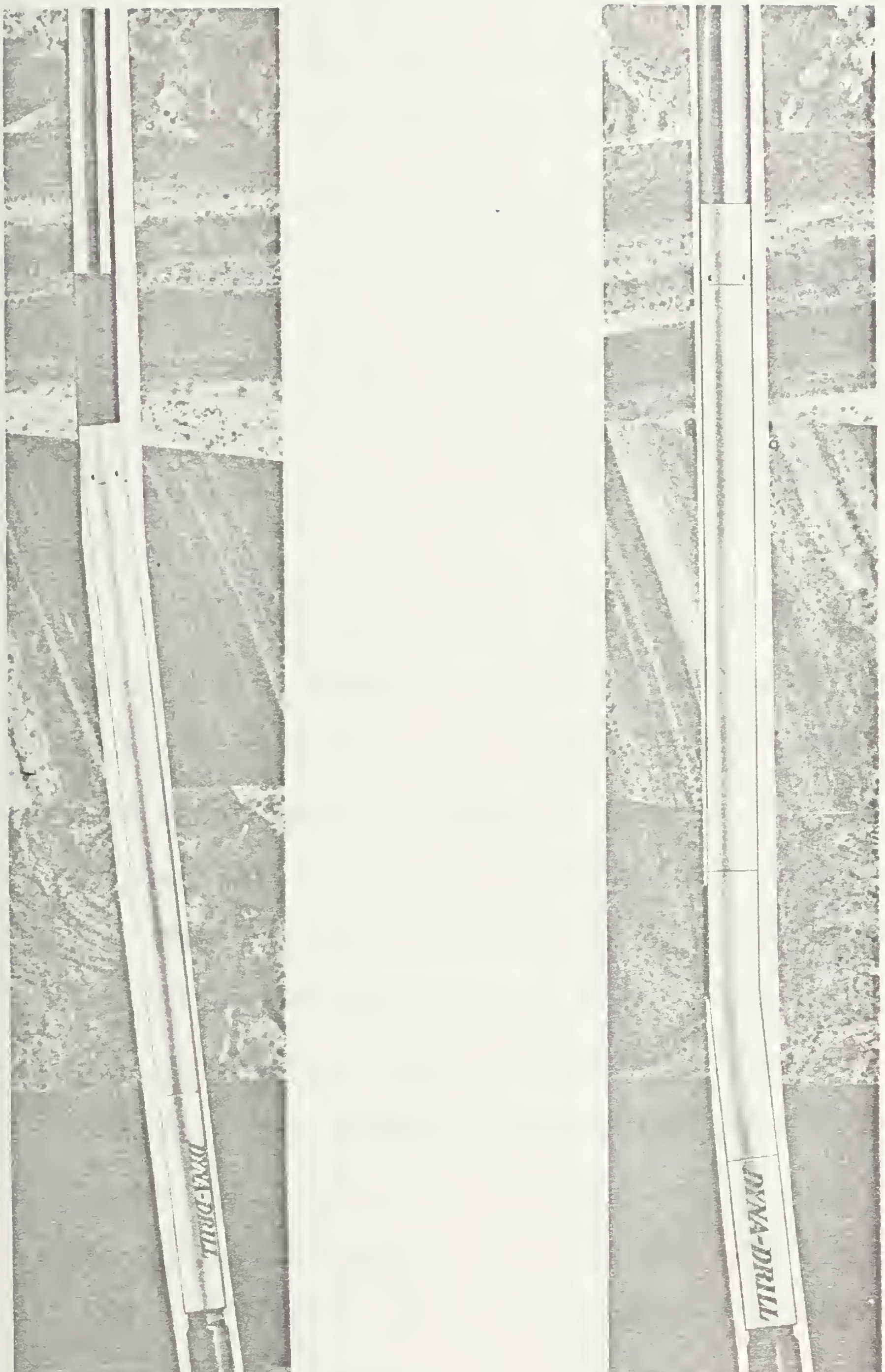


FIGURE C.6 Dyna-Drill Bent and Straight Housing Assembly

Two sizes of the Dyna-Drill have been considered in the final equipment design. The specifications for these two devices are listed in Table C.1

Table C.1 Dyna-Drill Specifications

O.D. (in)	Length (ft)	ΔP (psi)	RPM	GPM	HP	Wt (lb)	Torque (ft-lb)	Hole Size (in)
2-3/8	7	600	1000	25	6	60	30	4-1/2
6-1/2	19.6	250	305	250	28	1422	467	12

C.3 TURBINE DRILL

In 1960, directional drilling companies were beginning to appear in the oil well industry with a turbine downhole motor attached to a bent sub. This motor made it possible to deflect a well bore without the use of a conventional whipstock.

The turbine motor has three sections to it: a turbine section shown in Figure C.7; a replaceable bearing section shown in Figure C.8; and a rotating bit sub. The turbine section contains bladelike rotors and stators illustrated in Figure C.9. The stator is attached to the outer casing and is held stationary, while the rotor is attached to the shaft. Each rotor/stator section is called a stage. Several stages are combined to make a turbine.

As the drilling mud is pumped down through the turbine, the stator blades direct the fluid into the

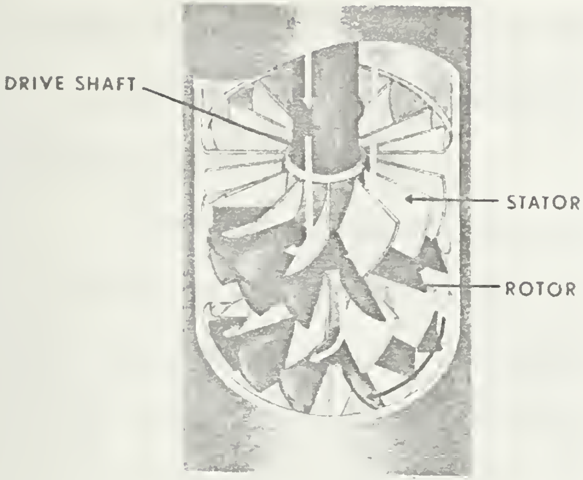


FIGURE C.7 Turbine Section of Turbo-Drill
(After U.of Texas, 1972)

FIGURE C.8 Replaceable Bearing Section
(After Eastman, 1969)

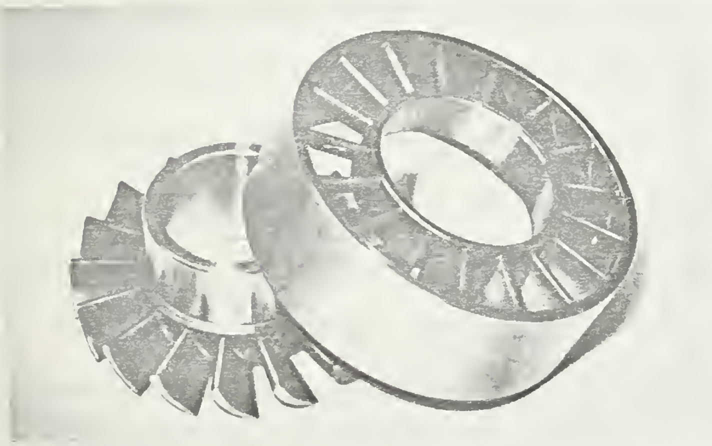


FIGURE C.9 Turbine Rotor and Stator
(After Eastman, 1969)

rotor blades. The flow of the drilling mud forces the rotor to rotate the shaft clockwise, which in turn drives the bit.

Several difficulties arose in the initial application of a turbine motor to directional drilling. The operating rpm of this motor is approximately 1000 rpm. Therefore, the only bit which could be successfully used with it was a diamond bit, while the high rpm greatly reduced the bearing life. The high rpm's and relatively low horsepower output tended to result in the turbine motor stalling in soft, sticky clay.

When the turbo-drill stalls, there is no direct indication of this condition as there was with the Dyna-Drill.

Finally, the turbo-drill is very sensitive to bending because of the required alignment between the rotor and stator blades. Any bending, resulting from a sharp build angle, would result in binding or excessive damage to the turbine.

For the reason stated above, along with the excessive weight and length of the turbo-drill, lead the writer to the conclusion that this downhole motor would not be considered in the final equipment design recommendations for horizontal drilling in soft ground. Table C.2 lists the various specifications

for two sizes of turbo -drills available from Eastman Whipstock, Incorporated. The cross-section of an Eastman turbo-drill is shown in Figure C.10.

Table C.2 Turbo-Drill

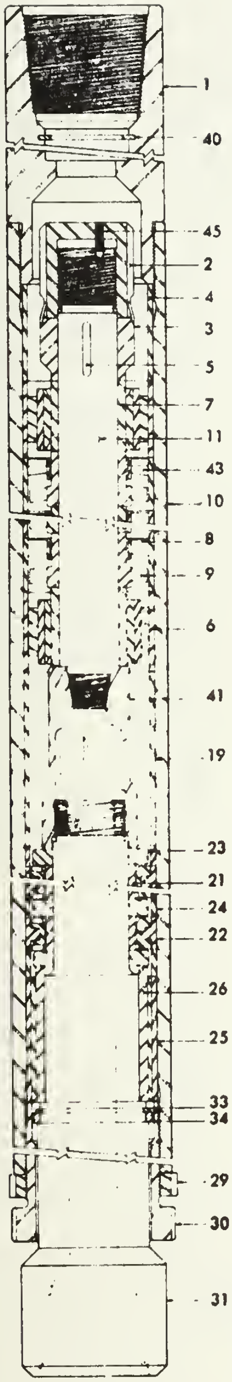
O.D. (in)	Lgth (ft)	Δ P (psi)	RPM	GPM	Stages	H.P. max	Wt (lb)	Stall Torque (ft-lb)	Hole Size (in)
$5\frac{1}{8}$	$18\frac{1}{2}$	328	780	250	60	26	750	272	$7\frac{7}{8}$
$6\frac{3}{4}$	$23\frac{3}{4}$	369	813	400	76	49.8	1985	591	$7\frac{7}{8}$

C.4 ELECTRIC MOTOR

Earth drilling with an electric motor is not a new idea. The first use of this method goes back to the early 1950's in Russia. However, the length of these electrodrills ranged from 36-42-1/2 ft(11-13 m) with power requirements ranging from 100-230 Kw.

The electric motor considered in this thesis for use with a thrust applicator is an order of magnitude smaller in size, while the power requirements are 10 orders of magnitude smaller than these Russian electrodrills. This is because the requirements for drilling small diameter holes in soft ground are minimal compared to that being required of an electro-drill in larger diameter holes.

Continental Oil Company (CONOCO) has successfully adapted an ordinary submersible pump motor to drill 6 in(15 cm) diameter holes in soft coal (Dahl, 1975).



DIRECTIONAL
INTEGRAL
TURBODRILL

Item No.	Description
1	Top Sub
2	Shaft Cap
3	Lockwasher-Turbine Section
4	Stator Spacer
5	Shaft Key-Turbine Section
6	Intermediate Bearing Body
7	Intermediate Bearing Sleeve
8	Stator
9	Rotor
10	Turbine Housing
11	Turbine Shaft
12	Internal Collar
17	Lockwasher-Bearing Section
18	Flow Ring
19	Spacer-Bearing Section
20	Shaft Key-Bearing Section
21	Thrust Bearing Sleeve
22	Thrust Bearing Body
23	Thrust Disc
24	Thrust Bearing Spacer
25	Lower Bearing Body
26	Lower Bearing Sleeve
28	Bearing Housing
29	Lower Sub Lock Ring
30	Lower Sub
31	Bearing Shaft
32	Lower Bearing Spacer
33	Retaining Ring
34	Catch Ring
36	Spline Clutch Box
37	Clutch Wear Pins
38	Clutch Spacer
39	Spline Clutch Pin Sub
40	Float Retainer Ring
41	Shaft Coupling
42	Assembly "O" Ring
43	Stator Screen
44	Lift Sub
45	Shaft Cap Lock Screw
46	Baker Float*

FIGURE C.10 Turbine Drill Cross-section (After Eastman, 1969)

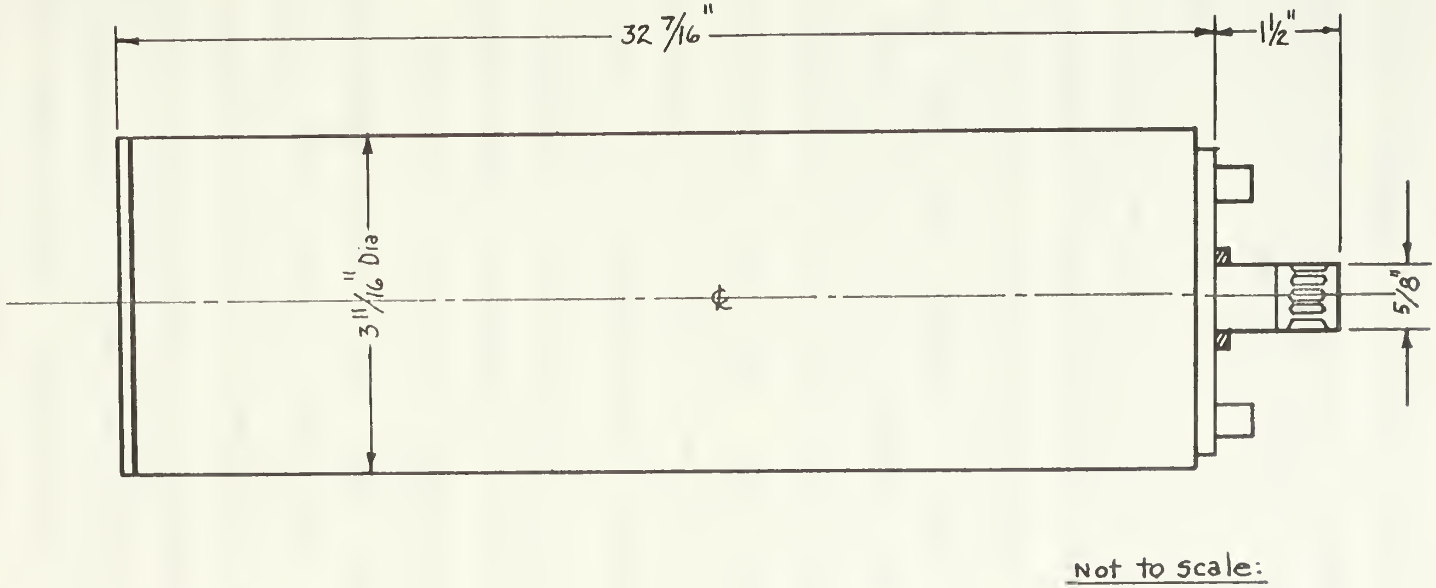


FIGURE C.11 Schematic of Century Electric Motor

This submersible pump motor is made by Century Electric Motor Company in Gettysburg, Ohio. The specifications for this motor are listed in Table C.3 while a schematic drawing is illustrated in Figure C.11.

The electric motor must be used with a reduction gear box because of the high output rpm's of the motor. A suitable planetary type gear box was designed by Reda Pump Company for one of their submersible motor pumps with available gear ratios varying from 28.51:1 to 3.165:1 for driving the drilling bit at 121-1095 rpm. The outside diameter of this unit is 4-1/2 in(11.4 cm) with a length of about 2 ft(0.6 m), wieghing 100 lbs(45.3 kg).

Two important considerations should be made if this electric motor is adapted to drilling in soft ground. Quick trip overload protectors should be used in all three legs of the three phase motor. This will prevent lock-up of the motor if it stalls from an overload and is not restarted immediately.

Another consideration is to maintain a constant flow of drilling fluid over the outside of the motor to prevent overloading. The standard flow requirement is a gallon/minute/H.P. which is easily satisfied by all of the final design models.

Table C.3 lists the important specifications for the Century submersible motor.

Table C.3 Century Electric Motor

O.D. (in)	Length (ft)	Voltage	Amps	HP	RPM	Wt (lb)	Hole Size (in)
3-11/16	2.7 (4.7 with gear box)	460	10.0	5	3450	50	7

C.5 HYDRAULIC MOTOR

In an attempt to find a stubby, small diameter, lightweight hydraulic motor that could be adapted to the DRILCO thrust applicator, Continental Oil Company (CONOCO) had the W. H. Nichols Company in Waltham, Massachusetts make a special order gerotor pump motor (Coffey, 1975).

The outcome of this special order was a gerotor, internal gear, positive displacement pump motor. This motor was designed for a flow rate of 30 GPM, producing 300 rpm at 10 H.P. output and operating at 75% efficiency. Any type of drilling fluid can be used to drive this motor. Because the torque vibrations of the gerotor are minimal, there is essentially no vibration associated with the operation of this motor.

The most important part of this motor is the gerotor in Figure C.12. The gerotor element consists of an inner and outer gerotor and an eccentric

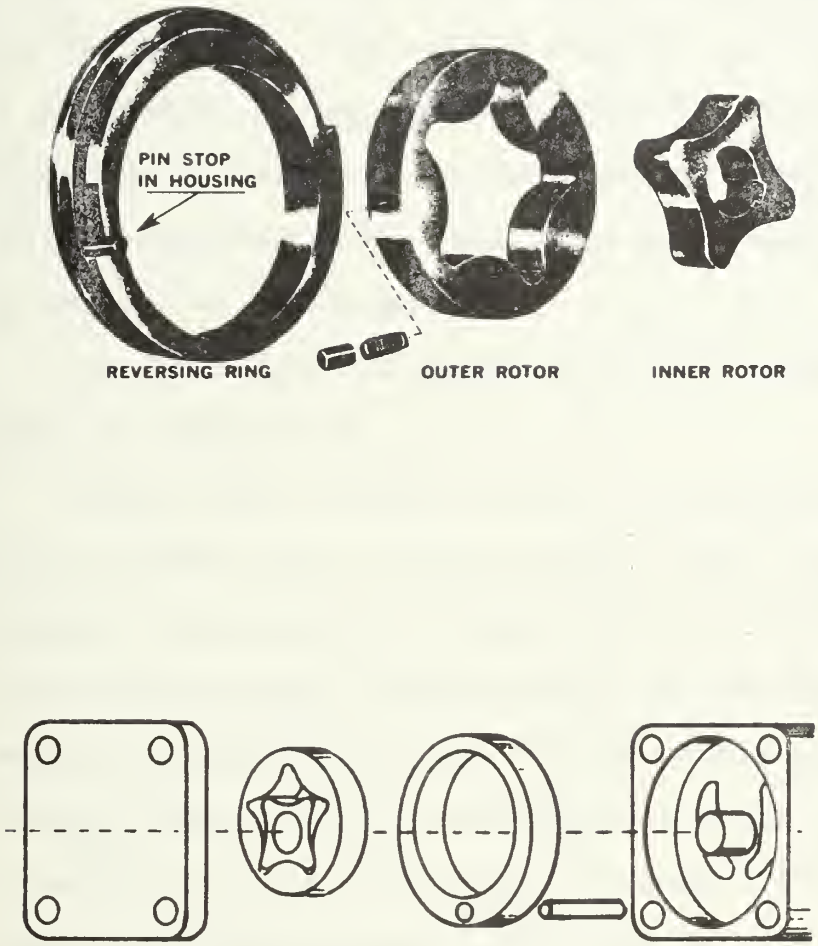


FIGURE C.12 Gerotor (After Nichols)

locator-ring. The inner gerotor always has one tooth less than the outer gerotor. This unit is placed in a casing or frame which provides housing and porting for the gerotor. The output capacity of this motor is a function of the number of gerotor units that are connected in series. For the previously mentioned design characteristics, the final motor had 16 sets of gerotor units in series.

The principle of operation of the gerotor is shown in Figure C.13.

Inlet ports in step 1 allow drilling fluid to fill a volume equal to the missing tooth. The toothed elements are mounted on fixed centers but turn eccentric to each other with the inner gerotor being mounted to the drive shaft. As the gerotors turn through steps 2 and 3 the chamber in which the fluid is carried decreases in size. At step 4 the fluid is forced out the discharge port into the next gerotor in series.

Table C.4 lists the important specifications for the particular motor designed for CONOCO.

Table C.4 Hydraulic Motor

O.D. (in)	Length (ft)	ΔP (psi)	RPM	GPM	HP	Wt (lb)	Torque (ft-lb)	Hole Size (in)
5	4	570	300	30	10	25	175	7

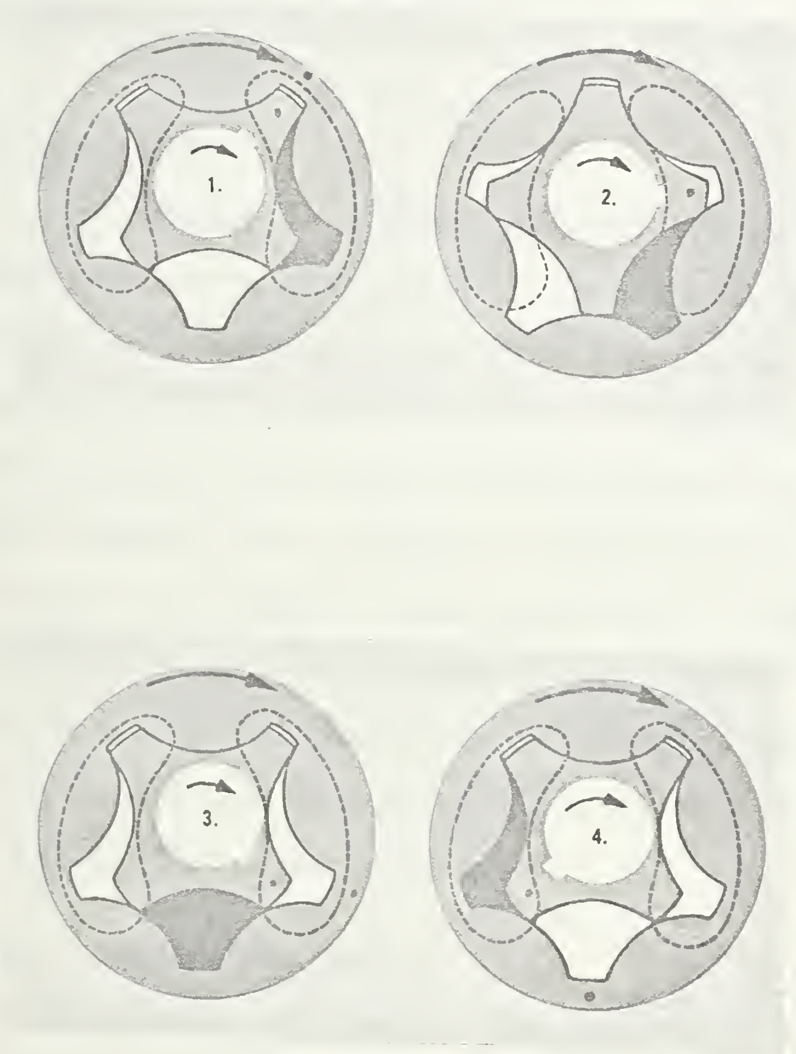


FIGURE C.13 Principle of Operation of Gerotor

APPENDIX D

SPECIFICATIONS ON DOWNHOLE THRUST APPLICATORS

D.1 INTRODUCTION

A thruster is defined as a relatively short device which functions solely downhole by providing a base for reactive torque from a drilling motor or for a reactive normal force from a compacting mechanism which in turn forms a subterranean hole. This appendix will elaborate upon four of the devices: the DRILCO thrust applicator, NURAT, U.S. Navy Polytordial Tunneler, and WORM TM*. If additional information is required by the reader, he is encouraged to contact the specific individuals on the List of Contributors.

D.2 DRILCO THRUST APPLICATOR

The thrust applicator manufactured by DRILCO, in Midland, Texas is a fully developed and operational thruster invented by Jack Kellner. This device can load and advance any type of drilling motor in any direction. Most of its application to date has been

* The name WORM is the trade mark which the inventor intends to apply to this system. It is so identified to preclude its assuming a generic connotation (Still, 1975).

in horizontal drilling, primarily in coal, with the longest hole bieng 1000 ft(305 m) at a diameter of 6 in(15.2 cm). Figure D.1 is the 2-3/4 in(7.0 cm) version of the thruster laying beside a 1-3/4 in (4.5 cm) Dyna-Drill.

A schematic of the thrust applicator in Figure D.2 will be helpful in explaining the operation of this device. The thruster is a double-acting cylinder having a hollow piston rod running through both ends. There are two anchor positions; the cylinder anchors, and the piston rod anchors. The anchor pads shown in Figure D.3 are made of steel with cross ribbing to improve their frictional characteristics. The unit shown in Figure D.3 is a complete anchor set for the 5-3/4 in(14.5 cm) which has three anchor pads positioned at 120° intervals around the sleeve. The dark area next to the anchor pad, in Figure D.3, is a hard elastic rubber which is molded to the metal body, including the entire internal circumference of the anchor sleeve. This rubber serves two purposes. First it provides a means for returning the anchor pad to its original position after the hydraulic pressure is released. It also eliminates the problem of any particles being caught underneath the anchor pads as they contract for repositioning. As many as three sets of anchor

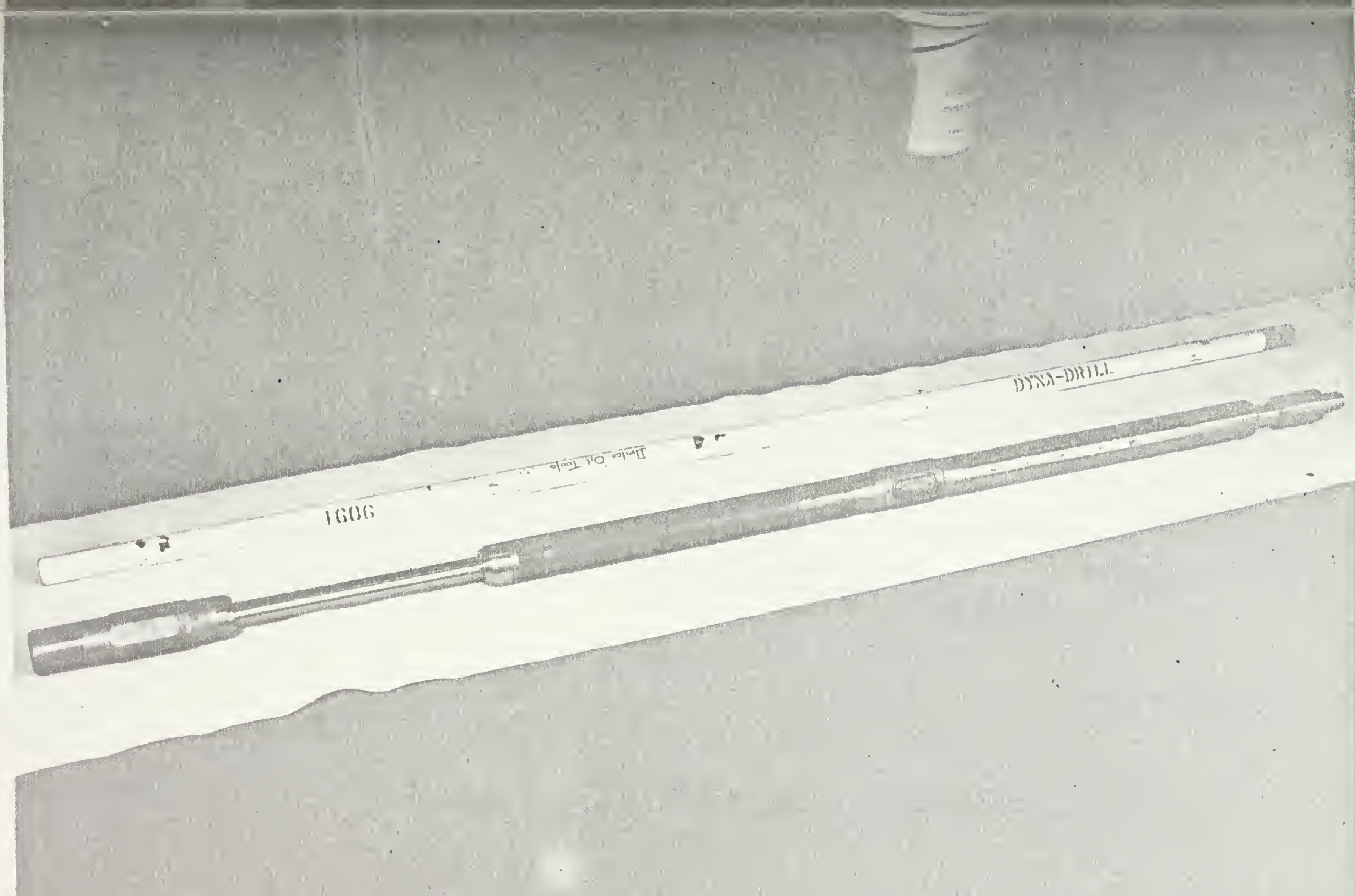


FIGURE D.1 DRILCO Thrust Appoicator System

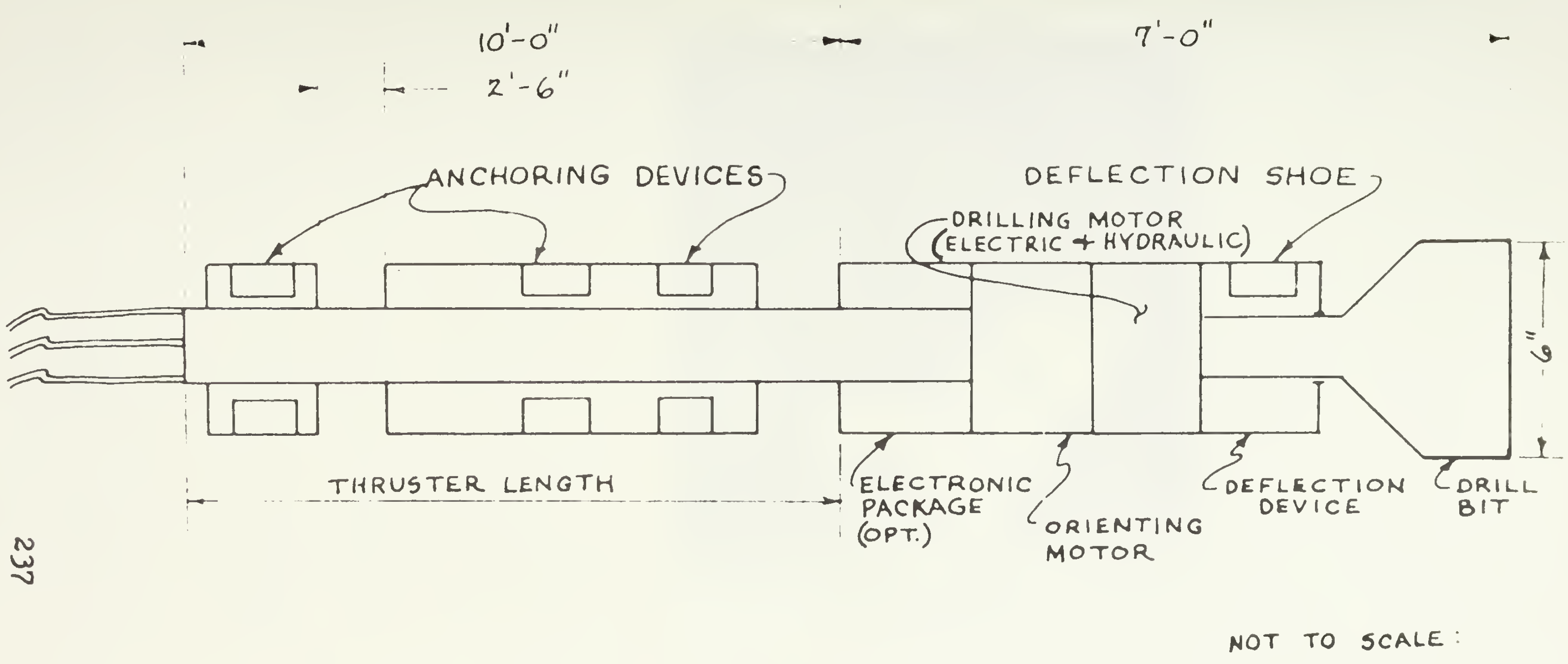


FIGURE D.2 Schematic of Thrust Applicator

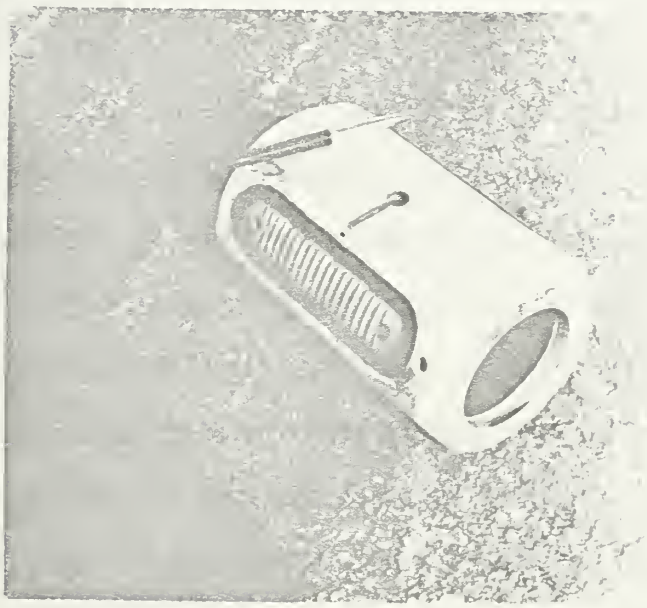


FIGURE D.3 DRILCO Anchor Pad

units can be attached to the thruster in the cylinder anchor section, while at the present time only one set of anchor pads can be attached at the piston anchor section.

In order to prevent rotation, there is a splime between the extension piston rod and anchor cylinder section.

The operation sequence of this thruster is as follows: (1) pressure is applied to the cylinder anchors, securing them to the drill hole wall; (2) pressure is then applied to the "out-hole" piston which moves the piston rod forward, thereby providing forward thrust to the drilling motor as it advances in the hole (the advance is limited by the stroke of the device); (3) at the end of the stroke the cylinder must be reset, therefore the piston rod anchors are set against the drill hole wall; (4) next pressure is released from the cylinder anchors which retract; (5) pressure is then applied to the "in-hole" side of the piston, forcing the cylinder toward the bit one stroke length. The thruster is then in position for another stroke. The hydraulic power unit is designed so that the resetting operation is done automatically in 15-20 seconds (Kellner, 1974).

The auxillary equipment located on the surface for this thrust applicator include 3-5 hoses which

are attached to the rear of the device, a 5 H.P. hydraulic power unit, and a means for powering the drilling motor which can be either hydraulic (water or mud) downhole motor, modified hydraulic motor, or an electric motor. Figure D.4 illustrates the surface equipment setup and required operating personnel. This picture was taken at the DRILCO test site in Midland, Texas.

Presently, DRILCO has the ability and experience to produce a thrust applicator which would be more compatible to soft ground operation than the current models. Future research and development for DRILCO in this area will obviously be a function of the market's demand for this thrust applicator.

Table D.1 lists the important specifications for the thrust applicator.

Table D.1 DRILCO Thrust Applicator

O.D. (in)	Length (ft)	Stroke (in)	Weight (lb)	Hole Size (in)
2-3/4	7.6	18	80	3-1/8
5-3/4	10.6	30	200	6

The primary user of this thrust applicator has been the Continental Oil Company(CONOCO). As a result of their research program, a deflection shoe and orientating motor have been developed and

FIGURE D.4 Operational Test of
DRILCO Thrust Appoicator

successfully tested in combination with the thrust applicator.

The deflection shoe and orientating motor are shown in their respective locations in Figure D.2. The deflection shoe serves two purposes: (1) directional control device which applies a lateral force on the bit, (2) correction device for precession of the thruster. The function of the deflection shoe as a directional control device is thoroughly explained in Chapter 3. The function of the deflection shoe as a correction device for precession results from the thruster inherently precessing $1/2^{\circ}$ per stroke because of splime error (Edmond, 1975). The deflection shoe is fully extended under 600 psi($4140 kN/m^2$) and requires a constant pressure of 220 psi(1518 kN/m^2) to maintain a constant elevation on a horizontal drill path.

The orientating motor is hydraulically operated in two basic modes. The first mode is a slow pulse which orients the deflection and a pulsated signal to correct for precession. The orientating motor initially rolls at 7° increments which then settles back to a total angular change of 4° where it positively locks into position.

One of the most significant improvements to the thrust applicator made by CONOCO has been the

reduction of hydraulic lines from 5 to 3 by adding a downhole valving system. The valving system is a set of hydraulic control valves with one control valve for the thruster and the other one for the orientating motor-deflection shoe circuit. Both of these systems vent the fluid to the annulus, thereby completing the open hydraulic system. A future development will be to have a completely closed hydraulic system which would mean having only one hose for the slurry line with a control line or hose within this slurry hose for control of the closed hydraulic system. The total resistance due to the normal weight friction component will then be decreased, thereby increasing the operational distance of the thrust applicator.

CONOCO has been able to drill a 6 in(15.2 cm) diameter hole 1000 ft(305 m) horizontally in soft coal. They have also been able to use three cylinder anchor sets and operate in 1 tsf(96kN/m^2) soft coal using the 5-3/4 in(14.6 cm) O.D. thrust applicator.

D.3 NURAT

NURAT is an acronym for Newcastle University Root Analogue Tunneller. This particular device was the result of research performed by Dr. D. Hettiaratchi, lecturer in the Department of

Agricultural Engineering at The University of Newcastle Upon Tyne, Newcastle Upon Tyne, England. No detailed information could be released by the sponsoring agency, British Gas Corporation, as a result of patents pending.

The device is not only a thruster, but also penetrates the soil with its cone-shaped front piece. The principle of operation for NURAT is based on Dr. Hettiaratchi's research of the mechanism associated with root penetration in dense soil. "The tunnel is formed by expansion of a hole from zero radius (Hettiaratchi, 1974)." As the anchor pads are extended radially outward, stress relief occurs at the tip of the device, thus allowing it to penetrate out ahead of the main body, compacting the soil around the cone.

The prototype of NURAT is approximately 3.3 ft (1 m) long and creates a hole about 6 in(15 cm) in diameter. The penetration rate of this prototype device was about 20 ft(6 m) per hour. There was no directional control device for the NURAT prototype.

The production development and future research programs involving NURAT have been passed on to the British Gas Corporation. The present developmental work being conducted by the British Gas Corporation

is geared to meet the following general design specifications (Spearman, 1974), stated in Table D.2.

Table D.2 Design Specifications for NURAT

O.D. (in)	Length (ft)	Weight	Power Source	Rate of Penetration	Comments
6	5	Capable of being handled by two persons	Mobile Hydraulic Power Pack	60 ft/hr in sand or clay	Should have ability to reverse direction

D.4 U.S. NAVY POLYTOROIDAL TUNNELING THRUSTER

The Naval Civil Engineering Laboratory at Port Hueneme, California conducted a feasibility study of vermiculating or wormlike motion as applied to a thrust device. The objective of this study was to determine the feasibility of a polytoroidal tunneling thruster concept, especially its application as a thrust device for penetrating rock, clay, or sand in combination with a drilling motor or direct displacement method of penetration. High thrust was developed by using large contact surfaces while axial movement was provided by means of a vermiculating motion. Vermiculation is "a motion in which a longitudinal wave traverses a contacting surface in the direction of translation (Williams and Gaberson, 1973)."

The basic idea for a polytoroidal tunneling thruster evolved from careful study of the tunnel boring machines. It was noted that these devices lacked the versatility to operate in both hard rock and soft ground. Therefore, the theory behind the polytoroidal tunneler applies large contact surfaces, using low operating pressures, in order to provide a high thrust capability. The model used to test this theory is shown in Figure D.5.

The toroids squeeze against the tunnel wall and remain in position due to the frictional characteristics of the soil media. The thrust provided by each toroid was calculated using, $P=\pi Dwp\mu$, where D is the toroid diameter, w is the surface contact width, p is the inflation pressure, and μ is the coefficient of friction of the soil. This is the same relationship as the Mohr-Coulomb failure criteria (i.e. $\tau_{ff}=\bar{\sigma}_1\tan\bar{\phi}$) for cohesionless soils. If both sides of the Mohr-Coulomb equation were divided by the contact area, then the resulting force would be maximum thrust available from the thrust device.

The operation of the polytoroidal thruster is illustrated in Figure D.6. In step (a) the most forward toroid is deflated, in step (b) the device has advanced one step because the forward bag expended simultaneously while the after bag deflated.

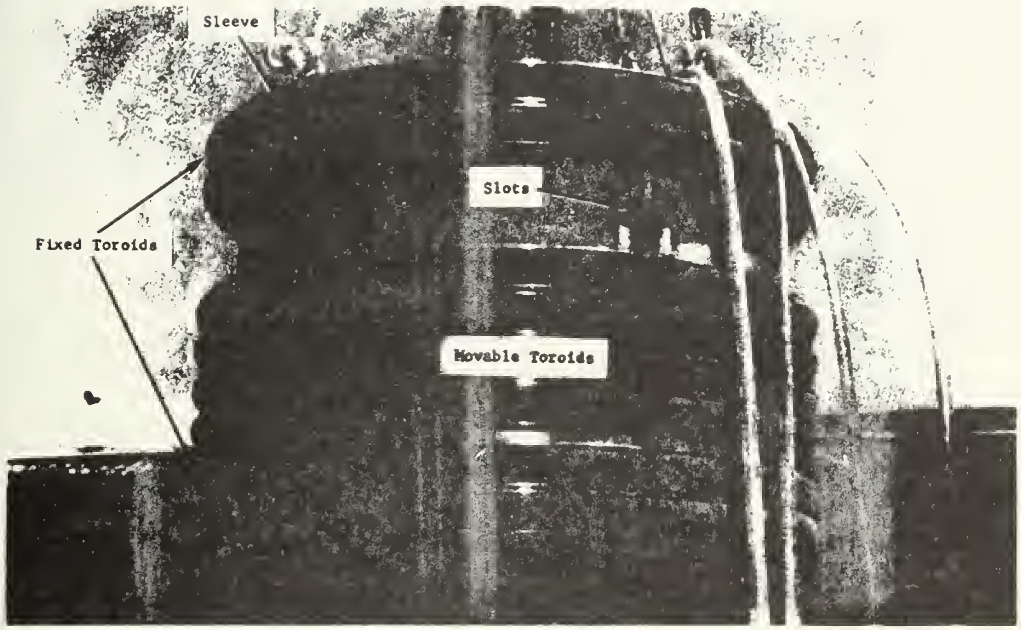


FIGURE D.5 U.S. Navy Polytoroidal Tunneling Thruster (After U.S. Navy, 1973)

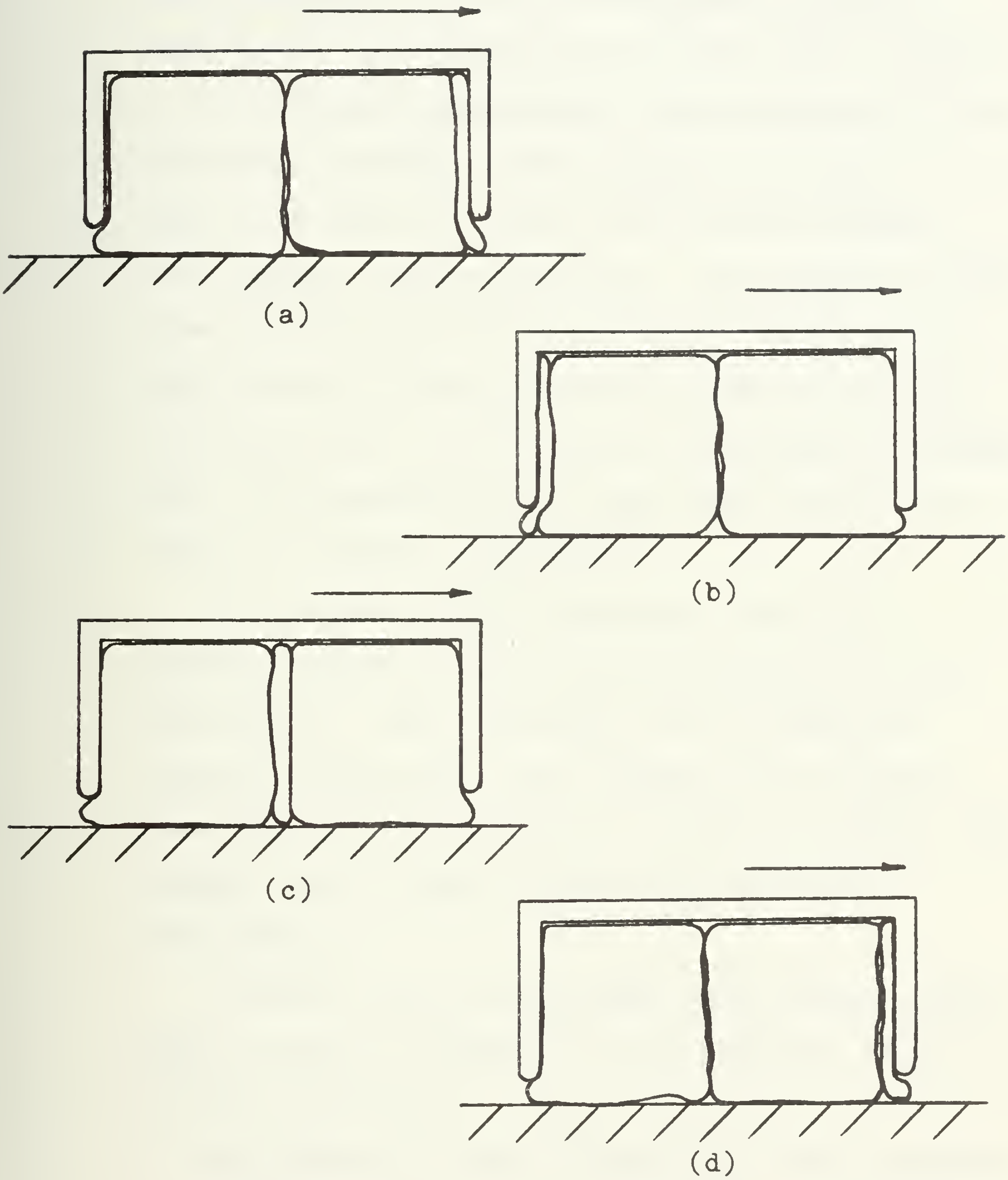


FIGURE D.6 Operational Sequence of the Polytoroidal Thruster (After U.S. Navy, 1973)

In step (c) the middle bag deflated while the after bag was inflating. Finally, in step (d) the forward bag is deflated. This then completes the cycle. With each step described the thruster moves forward.

Preliminary experimental tests verified that the theory was feasible. These results initiated a search for material to make the toroids stronger, more durable, and more flexible. The internal working pressure was set at 10-50 psi(69-315 kN/m^2) while other design criteria included a cyclic inflating/deflating life of 10000 cycles, low weight-to-strength ratio, low permeability to gases, and a high resistance to an adverse environment.

The result of this industrial search for a suitable toroid concluded that bladders had to be custom made. "The technology and the materials required to fabricate such bladders are available commercially (Pal and Gaberson, 1974)." However, the purchase cost of these bladders was considered noneconomical.

A second test was performed using bicycle inner tubes as shown in Figure D.5 The model was able to lift 600 lb(272 kg).

The results of their feasibility study indicated that the polytoroidal tunneler is a very reasonable method of applying thrust to a downhole motor.

For example, an 8 ft(2.4 m) device was axially tested with a thrust of 55000 lb(540 N) with an air pressure of 50 psi. However, due to the high cost for developing this device, the Naval Civil Engineering Laboratory was unable to continue its research for this project.

D.5 WORM

WORM is an acronym for Wheel-less Orthogonal Reaction Motor which is a downhole drilling system as shown in Figure D.7 and was invented by William Still from Aerospace Industrial Associates, Incorporated. This design approach solves two major and costly problems in drilling horizontally at long distances. First, like the previous thrusters, it provides a constant force at the bit, independent of the distance along the drill hole, and secondly it provides adequate maneuverability and orientation within the system so that it can function continuously without stopping for a survey (Still, 1975).

Presently, the WORM is still in the embryonic stages of development and the principle of operation has only been tested with a small model. No prototype has been built or tested ina subsurface environment, to the best of the author's knowledge.

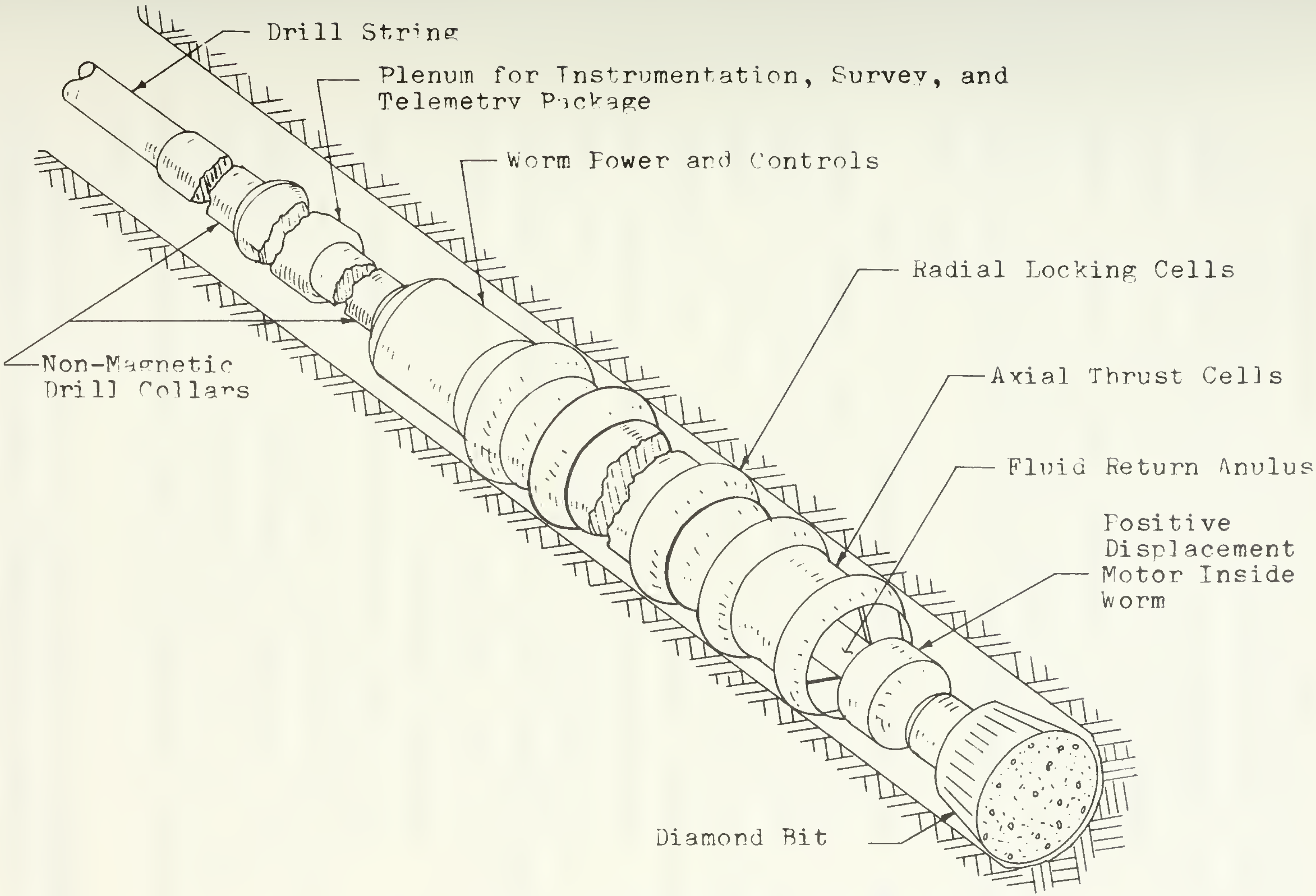
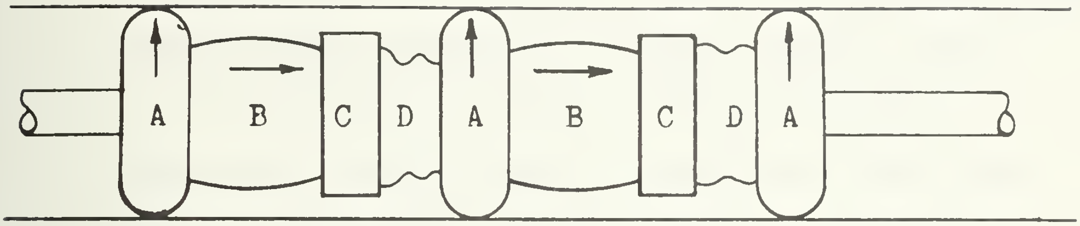


FIGURE D.7 WORM™ (After Still, 1975)

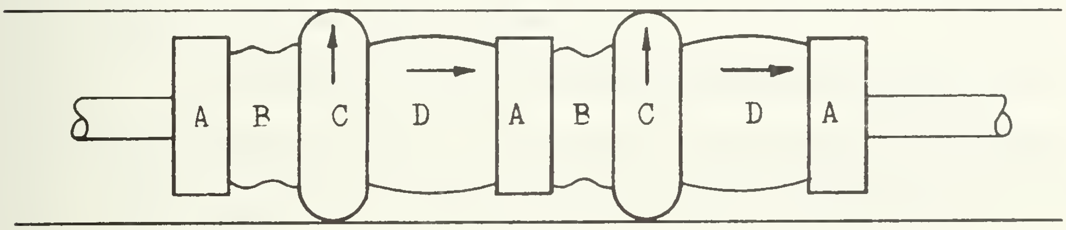
The major design difference between the WORM and a thrust applicator is the replacement of the conventional individual anchor pads with elastomeric cells, as shown in Figure D.7. This then is an advanced form of the same concept presented in the section on the U.S. Navy Polytoroidal Tunneler. The locomotion principle applied here is the vermiculating or worm-like motion.

Figure D.8 is an explanation of exactly how WORM propells itself down the hole. The cells shown in Figure D.8 are what the inventor calls, "vector force cells." There are two types of these cells, an axial and radial cell. The axial cell expands and applies force parallel to the axis of the borehole with insignificant radial expansion. The radial cells then expand radially outward from the borehole axis to provide contact surface for anchoring. The choice of an elastomer for the cell material allows for cyclic expansion without excessive damage to the drill hole wall. In addition, the properties of the elastomer, such as high abrasion resistance, high strength, and its incompressability, make it a very desirable material for use in a subterranean environment (Still, 1975).

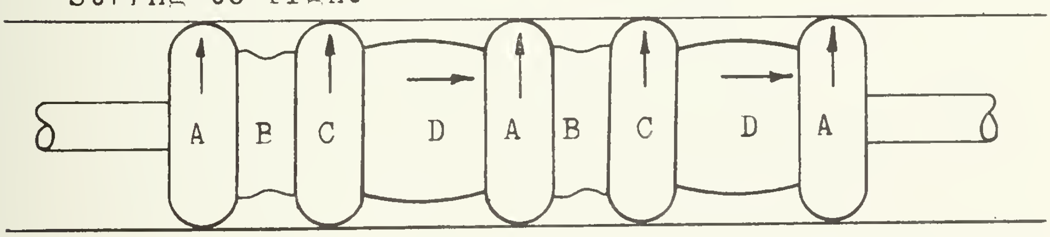
Directional control of this device is accomplished by controlling the degree of parallelism between the



"A" Expands, locking drill string at these points
"B" Expands, forcing "C" to right and compressing "D"
"C" & "D" have been released from that shown in 3 below



"C" Expands, locking drill string at these points through "B" & "D"
"A" & "B" are released and "D" espands forcing "C" and "A" apart
"A" is attached to drill string, "C" is not. "C" is locked to wall of drill hole. Thus "A" forces drill string to right



"A" Expands, relocking drill string at new point cycles, then repeats to 1 above.

Note: Single action, which provides one thrust per cycle is the simplest; other actions which provide smooth power flow have been derived.

FIGURE D.8 Operational Sequence of the WORM (After Still, 1975)

various muscle units. "A controlled lack of parallelism will force the WORM body to swing into an arc of fixed radius of curvature (Still, 1975)."

The WORM, as shown in Figure D.7 has a hydraulic drilling motor to power the drill bit. Also shown in this figure, directly behind the motor, are annulus openings for the drilling fluid to return through the WORM unit and exit out the up hole end of the body to provide lubricity within the drill hole for the drill pipe.

The WORM concept has many positive aspects to it, however since it has not been built as a prototype and field tested, it was not considered in the final equipment design.

APPENDIX E

DIMENSIONLESS ANALYSIS CALCULATIONS

E.1 INTRODUCTION

A review of the literature and several communications with key personnel in the horizontal directionally controlled drilling industry revealed that no system of comparison or correlation existed for the various drilling systems available.
One major reason given for this state was that the variability of each drilling situation does not lend itself to a simple dimensionless ratio. Another reason was the difficulty in developing a set of parameters that, first would be meaningful, and second, be practical. Therefore, the selection of comparative parameters was tempered by the dual requirement of applicability in a variety of geological conditions and simple practicality.

Of the four parameters presented, three are dimensionless while the fourth one has units which are not significant for comparison. The dimensionless ratios are the shearing parameter, jetting parameter, and the fluid system parameter.

The dimensional parameter is the drill motor parameter.

Each of these parameters will be presented separately along with the logic of their derivation and the criteria necessary to evaluate a system with them. In addition, a sample calculation will be followed by a table which includes all of the values for the four systems selected. The reasons for evaluating these particular four systems are explained in Chapter 4.

E.2 SHEARING PARAMETER

The shearing parameter has been developed to indicate some measure of the torque required to fail the soil at the outer edge of the bit face, in relation to the torque that is available from a particular motor with a specific size drill bit.

The torque required to shear the soil was derived from the cylindrical torque equation with the maximum torque resulting at the bit-~~drill~~ hole wall interface.

$$T = \frac{\tau_{max}}{r} J$$

$\tau_{max} = S_u$ = undrained shear strength of the soil

r = radius at the bit-soil interface

$J = \frac{\pi d^4}{32}$ = polar moment of inertia

The resulting parameter is:

$$SP = \frac{\frac{S_u (J)}{r}}{T} = \frac{S_u \frac{\pi d^3}{16}}{T}$$

Two different undrained shear strengths were adopted for these calculations. An S_u=0.25 tsf (soft clay) is the best condition for shearing because of its low resistive shear strength while conversely, an S_u=2.0 tsf (stiff clay) is the worst soft ground condition with respect to shearing at the outer edge of a bit face.

The following is a sample calculation for a 2-3/8 in(6 cm) Dyna-Drill motor in a 4-1/2 in (11.4 cm) hole.

S_u=500psf

d=4-1/2 in=0.375 ft

Torque=30 ft-lb

$$S.P. = \frac{S_u \frac{\pi d^3}{16}}{T} = \frac{500\,\pi (0.375)^3}{16(30)} = 0.173$$

The criteria established to evaluate the shearing parameter is that, if the ratio is less than one, the drill motor and bit will be able to shear

the soil from a dead start. A value greater than one does not mean that the motor/bit combination will not be able to drill in that specific soil, but instead that if the drilling operation depended solely on the shear ability of the system at the soil-bit interface, then the system could not drill.

Table E.1 indicates the various values for the shearing parameter for each system considered in the final equipment design.

Table E.1 Shearing Parameters

Drill System	System Torque (ft-lb)	Shearing Parameter	
		S_u=500 psf	S_u=4000 psf
2-3/8 in O.D. Dyna-Drill 4-1/2 in hole	30	0.173	1.38
6-1/2 in O.D. Dyna-Drill 12 in hole	467	0.21	1.68
5 in O.D. Hydraulic Motor 7 in hole	175	0.111	0.89
3-11/16 in O.D. Electric Motor 7 in hole	175 @ 150 RPM	0.111	0.89

E.3 JETTING PARAMETER

One of the most important considerations to account for when selecting a drilling system to bore a hole in soft ground is, whether, in fact, the soil in front of the bit is being eroded under a high velocity stream of drilling fluid from the bit orifice. Some degree of jetting is desirable in order to increase the efficiency of the drill bit, however, an excess of jetting will create a large cavity in front of the bit as explained in Chapter 3.

The jetting parameter(JP) represents the velocity of a fluid to cause erosion of a particular soil in comparison to the jet stream velocity emitting from the bit orifice, or $JP=V_e(448.8)/GPM/A_{B.O.}$ The erosion velocity has been taken for water and not drilling mud since no data was available for mud slurry. Therefore, the erosion velocity is probably lower than it would be for a drilling mud.
Another important assumption is the jet stream flows directly from the orifice to the borehole face. This is a conservative assumption, since the flow pattern is in reality a vortex and the vortex flow would increase the erosion effect at the bit face.

The erosion velocity data was found for a sand-gravel soil and a clay soil. The value adopted for the erosion of sand was taken from Leet and Judsen

(1971), Figure 11.16. The erosion velocity in this figure is for turbulent flow in a stream for a 7mm diameter particle and equal to 9.84 ft/sec(3.0m/sec). Several reasons for using this value include:
(1) there were no values available in the literature for the critical erosion velocity in sand (i.e. that required for initial movement of a particle of sand at the soil-liquid interface; (2) the actual flow within the bit face is, in fact, turbulent; and (3) the 7 mm diameter size particle is an average particle diameter for compacted, cemented sands. A value for the erosion velocity in clay was calculated and empirically derived in the literature. An empirical average value was taken from Graf (1971) and equal to 4.69 ft/sec(1.43 m/sec).

A sample calculation of the parameter follows:

For sand V_e=9.84 ft/sec

For a 7 in Tricone bit with a bit orifice diameter of 13/32 in:

$$\text{Orifice Area} = \frac{\pi d^2}{4} = \frac{\pi (13/32)^2}{4(144)} = 0.0009 \text{ ft}^2$$

$$JP = \frac{V_e(448.8)}{\frac{GPM}{A_{B.O.}}} = \frac{9.84(448.8)}{\frac{30}{0.0009}} = 0.132$$

The criteria used to evaluate this ratio is that if the velocity required for erosion is greater than the bit orifice velocity ($JP > 1$), no erosion at the bit face occurs. Therefore, the smaller JP is the more erosion occurs in front of the bit.

To date there is no maximum limit to how small this number can be before detrimental jetting of a cavity occurs in the area surrounding the bit. However, this parameter can be used to evaluate the relative effects of jetting between various bit/motor/flow rate combinations for different soil environments.

Table E.2 presents the results of the jetting parameter for the various bit sizes considered in the equipment designs. The diamond bit is not included because it does not have orifices but instead has fluid passages.

Table E.2 Jetting Parameter

Bit Type	O.D. (in)	GPM	Orifice Dia (in)	Orifice Area (ft^2)	JP	
					Sand	Clay
Tricone	7	30	13/32	0.0009	0.132	0.063
	12	325	5/8	0.0021	0.029	0.014
Drag	4-1/2	25	5/16	0.000533	0.094	0.045

E.4 DRILL MOTOR PARAMETER

The two previously discussed parameters have been dimensionless and have dealt with the geological aspects of drilling in soft ground. We will now turn our attention to the drill motor parameter which will deal with the equipment characteristics of each drilling motor. The parameter is not dimensionless, however, the dimensions that do result from this ratio are not meaningful to the analysis. What is meaningful, is that the horsepower output of the motor with respect to the size (i.e. volume) of the motor, is compared to the torque output. Therefore, the drill motor parameter(DMP)=H.P.(550)/Vol/Torque.

A more meaningful parameter for evaluating different motors in various soil conditions was presented by Dr. Neville G. W. Cook at the Third Congress of the International Society for Rock Mechanics (Cook and Harvey, 1974). Dr. Cook evaluated the efficiency of excavating in rock in terms of the specific energy of rockbreaking and the specific power, that is the power that can be delivered to a unit area of the working face. The specific energy is the energy consumption per unit volume of the original solid rock that was broken. The specific energy is a function of the type and condition of the rock, the strength, and the size of broken particles. Dr. Cook

and his associates had done previous studies to determine these values for different size particles. The relationship adopted here is R=3600P/S where P is the power delivered to the working face per unit area, S is the specific energy for the method used, and R is the rate of penetration along the tunnel axis. Therefore, each system was compared on the basis of its rate of penetration.

This type of comparison would have been adopted to this thesis, however, the specific energies of various soils are unknown. This then is the reason for adopting the drill motor parameter in the form DMP=H.P.(550)/V_m/Torque.

The following sample calculation is for a 2-3/8 in(6 cm) O.D. Dyna-Drill.

$$\text{Output H.P.}=6$$

$$\text{Vol}=\frac{\pi d^2(L)}{4} \quad \text{where } L=7 \text{ ft}$$

$$\text{Vol}=\frac{\pi(2.375)}{4(144)}(7)=0.215 \text{ ft}^3$$

$$\text{Torque}=30 \text{ ft-lb}$$

$$\text{DMP}=\frac{6(550)}{0.215(30)}=511.6$$

The evaluation criteria for this parameter is one which considers the smallest value of DMP to be the most efficient use of the volume of the motor for the rated design power and torque outputs.

Table E.3 lists the results of DMP calculations for the four proposed equipment designs.

Table E.3 Drill Motor Parameter

	H.P.	Volume (ft^3)	Torque (ft-lb)	DMP ($1/ft^3$-sec)
2-3/8 in O.D. Dyna-Drill	6	0.215	30	511.60
6-1/2 in O.D. Dyna-Drill	28	4.52	467	7.30
5 in O.D. Hydraulic Motor	10	0.79	175	39.78
3-11/16 in O.D. Electric Motor & Gear Box @ 150 RPM	5	0.348	175	45.2

E.5 FLUID SYSTEM PARAMETER

The final parameter relates the annular pressure in the drill hole to the hydraulic fracture gradient of the formation being drilled. The annular pressure is the hydraulic pressure required at the bit to push the drilling fluid back up and out of the hole.
A convenient method used to express this pressure in mud weight is the equivalent circulating density which is completely explained in Chapter 3 and Appendix B. The hydraulic fracturing gradient for this parameter has been taken from the thesis work performed by Hedberg (1975). Therefore, the final form of the fluid system parameter(FSP) is

ECD/Hyd. Frac. Grad.

The purpose of this parameter is to objectively calculate whether, in fact, the soil formation will fracture in the direction of the minor principle stress, resulting in loss of circulation of the drilling fluid.

A simple calculation of the FSP for a 2-3/8 in (6 cm) O.D. Dyna-Drill is as follows:

$$\text{ECD}=1.21\ \text{g/cm}^3\ (@\ 100\ \text{ft depth})$$

$$\text{Hyd. Frac. Grad.}=\begin{matrix}1.6\ \text{g/cm}^3 & @\ \text{clay}\\ 1.5\ \text{g/cm}^3 & @\ \text{sand}\end{matrix}\quad \text{(Hedberg, 1975)}$$

$$\text{FSP}=\frac{\text{ECD}}{\text{HFG}}=\frac{1.21}{1.6}=0.76\ @\ \text{clay below the water table}$$

The criteria for evaluating this parameter is such that the ECD is not greater than the hydraulic fracture gradient. If the ECD is greater, then loss of circulation occurs which could result in the drilling system becoming stuck in the hole.

Table E.4 lists all of the values of FSP for the various systems considered.

Table E.4 Fluid System Parameter

Motor Type	ECD (g/cm^3)	Fracture Gradient (g/cm^3)		FSP			
		Clay	Sand	Clay		Sand	
				Depth			
				100	500	100	500
$2\frac{3}{8}$ in O.D. Dyna-Drill	1.3	1.6	1.5	0.76	0.91	0.81	0.97
$6\frac{1}{2}$ in O.D. Dyna-Drill	1.12	1.6	1.5	0.73	0.88	0.78	0.94
$3\frac{11}{16}$ in O.D. Hydraulic Motor	1.13	1.6	1.5	0.73	0.89	0.78	0.95

Note: The electric motor was not considered because it only requires a minimum flow rate and pressure for cooling and voiding drill fluid from the drill hole.

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